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— POWER TRAIN —

Hybrid Computer Simulation of a  
Babcock & Wilcox Nuclear Power Plant

by

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Key Words: POWER TRAIN, Hybrid Computer, Power Plant  
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ABSTRACT

POWER TRAIN is a hybrid computer simulation of the normal and transient power operation of Babcock & Wilcox nuclear power plants. The simulation runs in real time and employs three digital computers and two analog computer consoles. POWER TRAIN includes models of all major components in both the primary and secondary loops of the plant from the reactor core to the turbine generator, including the pressurizer, steam generators, feedwater heaters, feedwater pumps, control valves, safety and relief valves, and the integrated control system.

Parameters describing plant components are input to the simulation in card image format. Pushbuttons on a "mini" control panel allow real-time operation of the simulation. Strip chart recorders are the principal output devices.

This report describes the computer facilities and the mathematical models used by POWER TRAIN. The description of the mathematical models includes modeling assumptions, equations, solution methods, limitations of the simulation, and a summary of the input requirements and output capabilities.

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## 1. INTRODUCTION

POWER TRAIN is a real-time, on-line hybrid computer simulation of power operation for Babcock & Wilcox nuclear power plants (177-, 205-, and 145-fuel assembly plants). The code is used to predict the performance and behavior of the major components in the nuclear steam system (NSS) for a wide range of plant conditions and operation. POWER TRAIN is designed to model as much of the power plant as is feasible, especially those components whose behavior is interrelated with that of others. This report describes the scope of the simulation, the modeling assumptions, and the modeling equations. Also given is an overview of the method of implementation, the input requirements, and the output capabilities.

The objectives of the simulation are listed below:

1. Two-loop simulation.
2. Automatic setup and checkout capability.
3. Flexibility to change system parameters, especially control system gains and setpoints.
4. Operation of the simulation (as nearly as possible) by actuating pushbuttons on or off.

The scope of the simulation, together with the automatic setup and checkout requirement, necessitated implementation techniques employing digital and hybrid (digital-analog) methods. The simulation required the resources of the two digital computers (CDC 1700 and EAI-640) and the two solid-state analog computers (EIA-680) that are part of the hybrid computation facility at the Nuclear Power Generation Division (NPGD) in Lynchburg, Virginia.

Calculational speed requirements and consideration of future machine requirements resulted in the use of a hardware floating point array processor (AP-120B) that performs calculations for the CDC-1700, and thereby greatly increases the amount of the simulation that can be programmed on the digital computers.

The requirement for flexibility in the simulation of the control system resulted in the application of Interactive Simulation Language<sup>1</sup>, which allows a digital computer to be programmed like an analog computer and allows convenient modification of control system parameters.

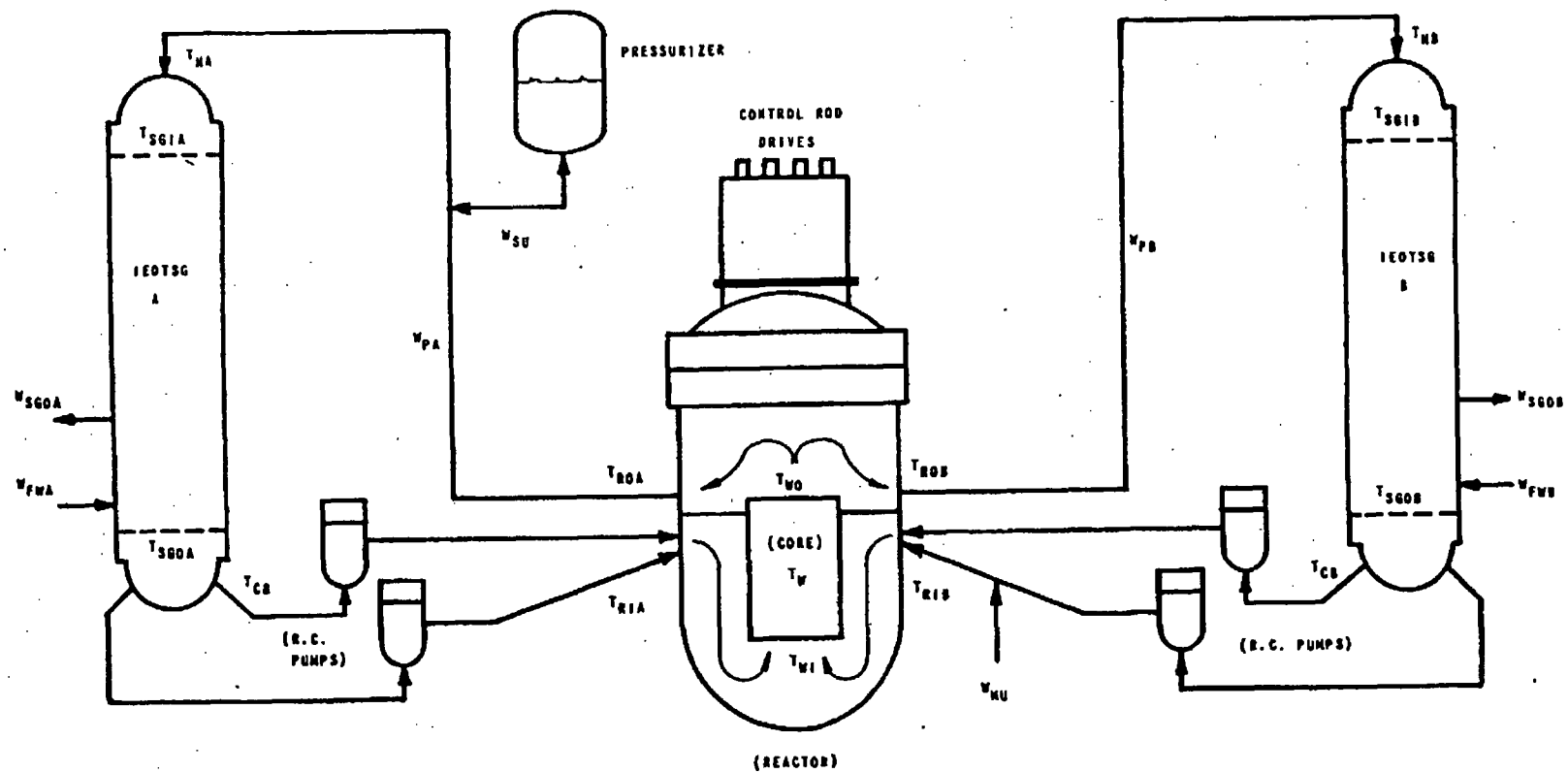
The desire for basically "hands-off" operation resulted in the specification of a "mini" control panel that allows automatic plant simulation but also allows the user interactive capability for controlling those portions of the simulation that are similar to a reactor operator's role of controlling an actual plant in the manual mode. In addition, this control panel allows the user to initiate many operational transients. The mini control panel was physically constructed on one of the analog computers and consists of on-off and momentary lighted pushbuttons. The status of all the pushbuttons is scanned by the EAI-640 computer; certain signals are then forwarded to the analog computers, while others are sent to the CDC-1700 via a digital data link between the digital computers..

Figures 1-1 and 1-2 are schematic diagrams of the power plant. With respect to these diagrams, the implementation of the major components on the various computers is as follows:

1. CDC-1700, AP-120B — Reactor, control rod drive, pressurizer, primary system flows and temperatures, condensate system flows, feedwater and condensate system energy balance, and turbine extraction flows.
2. EAI-640 — Control system (ICS), protection system, atmospheric dump valves, bypass valves, and steam relief valves.
3. EAI-680, Console 1 — Steam generators and portion of steam line.
4. EAI-680, Console 2 — Feedwater system flows, remainder of steam lines, feedwater control valves, feedwater pumps, and turbine-generator.

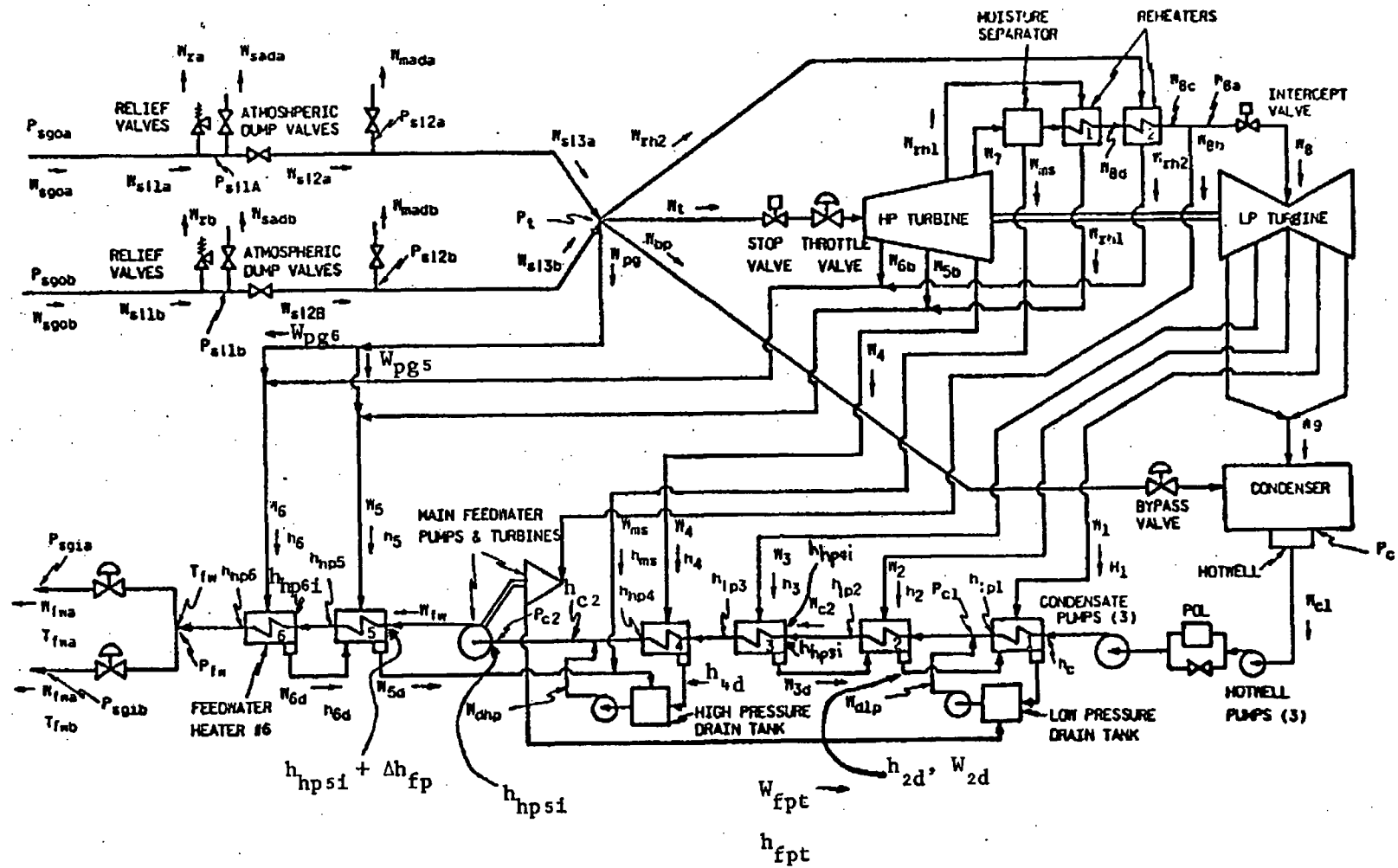
The analog computer portions of the simulation are set up and checked out automatically using COMANCHE, a program developed by Control Data Corporation for the CDC 1700 computer. COMANCHE is a system routine (resident on the CDC 1700) which sets potentiometers on the analog consoles and then performs a "static check" of the patchboards and analog components before performing production runs. All digital computer routines are either resident in the computer system or are read in from binary paper tapes.

Figure 1-1. Primary Loop Diagram



1-3

Figure 1-2. Secondary System Diagram



## 2. MATHEMATICAL MODEL

This section summarizes the equations employed by the POWER TRAIN simulation. Modeling assumptions are stated and the equations are presented with minimal discussion of the details of implementation. Solution methods are identified and pertinent derivations are given. Models are discussed on a component by component basis starting with components in the primary, continuing with the steam generator, and concluding with secondary components. The nomenclature for each subsection corresponds to the variable names used by the computer coding and are defined at the end of each section. This section concludes with a summary of the limitations of the simulation.

### 2.1. Scope of Nuclear Power Plant Simulation

The primary system of the nuclear power plant simulated in POWER TRAIN is diagrammed in Figure 1-1. The primary system model, when provided with appropriate input data, can be applied to any Babcock & Wilcox nuclear power plant (177-, 205-, or 145-fuel assembly). It comprises the reactor vessel, two steam generators, hot and cold leg piping, primary coolant pumps, the control and safety rod systems, and the pressurizer. The reactor vessel consists of inlet and outlet plenums and a reactor core with a bypass channel. Point reactor kinetics is used to model the dynamics of core power generation. The primary side of the steam generators consists of inlet and outlet plenums and a tube bundle in which the heat transfer process takes place. The pressurizer is modeled as a non-homogeneous thermodynamic system including the effects of heaters, sprays, wall condensation, and relief valves. Primary flow is modeled as a constant until on-off pushbuttons are actuated to simulate primary coolant pump failures. In those cases, primary flow is determined by flow versus time coastdown curves. All models, including the steam generators, can account for the effects of both forward and reverse flows. Reverse flow might occur if two primary pumps in one loop failed. All primary fluid is

assumed to be single-phase subcooled water except in the pressurizer. Instantaneous thermal mixing is assumed to occur in all plenums. Thermal transport in the piping is modeled with a variable transport delay.

The secondary system simulation is diagrammed in Figure 1-2. Included in the steam side of the secondary system are the steam lines connecting the steam generator to the turbine, turbine throttle, bypass and dump valves, and the turbine-generator. The feedwater and condensate side of the secondary system includes the condenser, hotwell, condensate, and feedwater pumps, three low-pressure heaters with a drain tank, three high-pressure heaters with a drain tank, and feedwater control valves for each steam generator. The feedwater system model uses a single feedwater train to represent two cross-coupled parallel trains of heaters and pumps. The arrangement of components in the feedwater system differs widely from plant to plant. The arrangement modeled by POWER TRAIN and shown in Figure 1-2 is based on the Tennessee Valley Authority (TVA) Bellefonte nuclear power plant. In addition, POWER TRAIN provides (as an option) the capability to select a variation of the feedwater system based on the Sacramento Municipal Utility District (SMUD) Rancho Seco nuclear power plant.

The steam in the steam lines is assumed to be of constant enthalpy. Conservation of mass and momentum equations are used to determine the flow and pressure responses in the steam lines.

The main feedwater flow is found by solving a momentum equation assuming constant density for the various sections of the feedwater piping.

The turbine extraction and feedwater heater drain mass flows are functions of throttle flow based on the steady-state plant heat balance information provided in the input data. The enthalpy of all extraction and feedwater heater outlet flows is also a function of throttle flow based on heat balance data. The dynamic behavior of these flows and enthalpies is modeled using a single time constant in a first-order differential equation. That is, when the throttle flow changes, these quantities will approach their new values asymptotically according to the time constant associated with each extraction flow and feedwater heater.

POWER TRAIN provides the option to model either a once-through steam generator (OTSG) or an integral economizer once-through steam generator (IEOTSG). The

OTSG requires modeling of a downcomer, a chamber in which incoming feedwater is preheated to saturation by recirculated steam before introduction to the tube bundle.

The integrated control system (ICS) is modeled extensively, including the unit load demand development subsystem, the integrated master control subsystem, the steam generator feedwater control subsystem, and the reactor control subsystem. In addition, the operator of the simulation may select, via the mini panel, either manual or automatic control of the following controllers and final control elements:

- Turbine bypass valves.
- Turbine throttle valves.
- Reactor-steam generator demand.
- Feedwater pump speed.
- Feed valve demand (A or B).
- Feed valve position (A or B).
- Reactor demand.
- Control rod position.
- Unit load demand.

Trips and abnormal conditions that may be initiated from the mini panel include the following:

- Manual reactor trip.
- Reactor coolant (RC) pump trips.
- Feedwater pump trips.
- Turbine trip.
- Loss of electrical load.
- Pressurizer heater failure.
- Enable/disable flux/flow trip of reactor.
- Enable/disable high primary pressure trip of reactor.
- Manual failure of drain and condensate pumps in feedwater train.

## 2.2. Reactor Kinetics and Reactivity Feedback

The point reactor kinetics equations with six groups of delayed neutrons are solved using the prompt jump approximation.

$$\rho = \frac{\rho^*}{\beta - \delta k_T} \sum_{i=1}^6 \lambda_i C_i \quad (2-1)$$

and

$$\frac{dC_i}{dt} = \frac{\beta_i}{\ell^*} n - \lambda_i C_i. \quad (2-2)$$

Reactivity feedback for fuel and moderator temperature changes are calculated using average feedback coefficients:

$$\delta k_{Dop} = \alpha_D [T_f - T_f(0)], \quad (2-3)$$

$$\delta k_{mod} = \alpha_m [T_w - T_w(0)]. \quad (2-4)$$

The effect of boron addition or dilution is modeled as a positive or negative rate of reactivity change:

$$\frac{d}{dt} \delta k_{bor} = \frac{R_{bor}}{3600} K_b, \quad (2-5)$$

$$K_b = \begin{matrix} 0 & \text{hold} \\ +1 & \text{borate} \\ -1 & \text{deborate.} \end{matrix} \quad (2-6)$$

Control and safety rod reactivities are obtained from the control and safety rod models.

The total reactivity is the sum of all these contributions. A shim reactivity is calculated by the simulation to obtain zero initial total reactivity:

$$\delta k_T = \delta k_{Dop} + \delta k_{mod} - \delta k_{bor} - \delta k_r - \delta k_{sr} - \delta k_{shim}. \quad (2-7)$$

#### Nomenclature

$C_i$	Precursor concentration of delayed neutrons in group i, neutrons/cm <sup>3</sup>
$\ell^*$	Mean effective prompt neutron lifetime, s
$n$	Neutron density, neutrons/cm <sup>3</sup>
$R_{bor}$	Boron reactivity addition (+) rate or dilution (-) rate, $\delta k/h$
$T_f$	Average fuel temperature, F
$T_f(0)$	Initial value of $T_f$ , F
$T_w$	Average temperature of water in core, F
$T_w(0)$	Initial value of $T_w$ , F
$\alpha_D$	Doppler coefficient, $\delta k/F$

$\alpha_m$	Moderator coefficient, $\delta k/F$
$\beta$	Fraction of fission neutrons which are delayed
$\beta_i$	Fraction of fission neutrons which are in delay group i
$\delta k_{bor}$	Negative reactivity due to boron addition, $-\delta k$
$\delta k_{Dop}$	Reactivity feedback due to fuel temperature, $\delta k$
$\delta k_{mod}$	Reactivity feedback due to moderator temperature, $\delta k$
$\delta k_r$	Negative reactivity due to rod insertion, $-\delta k$
$\delta k_{shim}$	Negative reactivity offset to give initial total reactivity of zero, $-\delta k$
$\delta k_{sr}$	Negative reactivity due to safety rod insertion, $-\delta k$
$\delta k_T$	Total reactivity, $\delta k$
$\lambda_i$	Decay constant for delayed neutron group i, $s^{-1}$

### 2.3. Control and Safety Rods

The control system transmits a signal to withdraw, hold, or insert control rods. The rod drive motor is simulated by two first-order differential equations for rod velocity and rod position.

$$\frac{dV_r}{dt} = (K_r V_r^* - V_r) / \tau_m, \quad (2-8)$$

$$\frac{dX_r}{dt} = V_r. \quad (2-9)$$

Control rod reactivity is calculated from linear interpolation of a rod worth versus rod position table included as part of the input data to the simulation:

$$\delta k_r = R_{rm} F_r (\%X_r) / 100. \quad (2-10)$$

Safety rod reactivity is obtained from linear interpolation over a table of a safety rod worth versus time after trip. The trip signal is initiated by the reactor protection system model. The table is included in the input data.

$$\delta k_{sr} = R_{sr} F_{sr} (t_{trip}) / 100 \quad (2-11)$$

### Nomenclature

$F_r$	Table of percent rod reactivity versus $\%X_r$ , %
$F_{sr}$	Table of percent safety rod insertion versus $t_{trip}$ , %
$K_r$	1 for rod insertion, 0 for hold, -1 for rod withdrawal
$R_{rm}$	Maximum total worth of control rods, $-\delta k$
$R_{srm}$	Maximum total worth of safety rods, $-\delta k$
$t_{trip}$	Time after signal to trip reactor, s
$V_r$	Rod velocity, in./s
$V_r^*$	Maximum normal rod speed, in./s
$X_r$	Length of rod inserted, in.
$\%X_r$	Percent of rod inserted, %
$\delta k_r$	Negative reactivity due to control rod insertion, $-\delta k$
$\delta k_{sr}$	Negative reactivity due to safety rod insertion, $-\delta k$
$\tau_m$	Rod drive time constant, s

### 2.4. Core Power

The core power is assumed to be directly proportional to the neutron density (section 2.2), provided that some minimum power value (POWMIN) specified in the input data is exceeded:

$$\%Q_f = n(\text{POWERI}) \quad \text{for } \%Q_f > \text{POWMIN.} \quad (2-12)$$

Below this value fission product decay is assumed to heat the fuel according to the following model. The power from the fuel decays exponentially with a single time constant ( $\tau_{dk}$ ) toward a non-zero asymptotic power level (DECAY).

$$\frac{d(\%Q_f)}{dt} = -(\%Q_f - \text{DECAY})/\tau_{dk} \quad \text{for } \%Q_f \leq \text{POWMIN} \quad (2-13)$$

### Nomenclature

DECAY	Asymptotic value of power for decay heat model, % of rated power
n	Neutron density expressed as a fraction of initial neutron density, dimensionless

POWERI	Initial value of core power, % of rated power
POWMIN	Threshold value of POWER below which the decay heat model is assumed to operate, % of rated power
%Q <sub>f</sub>	Power being generated in core expressed as a percentage of rated power, % of rated power
τ <sub>dk</sub>	Decay heat time constant, s

## 2.5. Core Heat Transfer

POWER TRAIN assumes that the average temperatures of the fuel, cladding, and cooling water may be determined from consideration of a single reactor control volume. Conservation of energy equations are written for the fuel, cladding, and coolant, where each region is modeled as a single node:

$$(MC_p)_f \frac{d}{dt} T_f = \%Q_f - (UA)_{fc}(T_f - T_c), \quad (2-14)$$

$$(MC_p)_c \frac{d}{dt} T_c = (UA)_{fc}(T_f - T_c) - (UA)_{cw}(T_c - T_w), \quad (2-15)$$

$$M_w C_{pw} \frac{d}{dt} T_w = (UA)_{cw}(T_c - T_w) - C_{pw} W_p (T_{wo} - T_{wi}). \quad (2-16)$$

Bender's method is used to calculate core outlet temperature, given the average water temperature<sup>2</sup>

$$M_w C_{pw} \frac{d}{dt} T_{wo} = (UA)_{cw}(T_c - T_w) - 2C_{pw} W_p (T_{wo} - T_w). \quad (2-17)$$

The heat transfer rate between cladding and coolant is assumed to be proportional to the 0.8 power of core coolant flow,

$$(UA)_{cw} = (UA)_{cw*} (W_p / W_{p*})^{0.8}. \quad (2-18)$$

The specific heat capacity of the core coolant is a polynomial fit with  $P_{rc}$  and  $T_w$  as independent variables:

$$C_{pw} = f(P_{rc}, T_w). \quad (2-19)$$

An initial core bypass flow rate may be specified via input to establish a ratio used to calculate core flow as a fraction of total primary system flow:

$$W_p = (W_{pa} + W_{pb}) \left[ 1 - \frac{W_{cbp*}}{W_{pa*} + W_{pb*}} \right] \quad (2-20)$$

### Nomenclature

$C_{pw}$	Specific heat of water, Btu/lbm-°F
$M_w$	Mass of water in the core, lbm
$(MC)_p^c$	Cladding mass times specific heat, Btu/°F
$(MC)_p^f$	Fuel mass times specific heat, Btu/°F
$P_{rc}$	Primary system pressure measured at pressure tap at surge line inlet, psia
$\%Q_f$	Power being generated in core expressed as a percentage of rated power, % of rated power
$T_c$	Average cladding temperature, F
$T_f$	Average fuel temperature, F
$T_w$	Average water temperature, F
$T_{wi}$	Core inlet temperature, F
$T_{wo}$	Core outlet temperature, F
$(UA)_{cw}$	Cladding-to-water heat transfer rate, % of rated power/°F
$(UA)_{cw}^*$	Initial cladding-to-water heat transfer rate, % of rated power/°F
$(UA)_{fc}$	Fuel-to-cladding heat transfer rate, % of rated power/°F
$W_{cbp}^*$	Initial core bypass flow rate, lbm/s
$W_p$	Flow rate through core, lbm/s
$W_{p}^*$	Initial value of $W_p$ , lbm/s
$W_{pa}$	Primary flow rate through loop A, lbm/s
$W_{pa}^*$	Initial value of $W_{pa}$ , lbm/s
$W_{pb}$	Primary flow rate through loop B, lbm/s
$W_{pb}^*$	Initial value of $W_{pb}$ , lbm/s

### 2.6. Primary System Flow Rates

The primary flow system is simulated as two loops connecting the reactor vessel with two steam generators (see Figure 1-1). In an actual B&W NSS there are two cold legs, each with its own RC pump. POWER TRAIN models these two

cold legs as a single cold leg with two pumps in parallel. As discussed in section 2.5, coolant flow through the reactor is modeled by dividing it into core and bypass flows.

The initial and normal primary flows (with all pumps operating) are constants specified in the input data. The capability to simulate four types of pump coastdown is provided:

- One-pump trip.
- Four-pump trip.
- Trip of two pumps in the same loop.
- Trip of two pumps, one in each loop.

Tables for flow versus time for each loop for each of the four cases are included in the input data. Reverse flow may be specified by negative values of flow. The direction of the flow will modify the calculation of loop temperatures appropriately.

## 2.7. Primary System Temperatures

Temperature response between any two successive points in the system is modeled by a flow-dependent mixing constant, a flow-dependent transport delay, or a set of differential equations characterizing the dynamics of the component between the two successive points.

The model used to calculate temperatures across plenums assumes that all inlet flow mixes instantaneously with fluid already in the plenum and that the outlet temperature is equal to the bulk average temperature. This model is used for the upper and lower plenums of the reactor and the steam generators. The form of the differential equation relating outlet temperature to inlet temperature is

$$\frac{dT_o}{dt} = \frac{\sum_{\text{all inlets}} (W_i T_i) - T_o \sum_{\text{all outlets}} W_o}{\rho V} \quad (2-21)$$

A flow-dependent transport delay model is used to calculate the temperature of the flow from the hot and cold legs of each loop. The expression for outlet temperature and the time delay is as follows.

$$T_o(t) = T_i(t - \tau) \quad (2-22)$$

$$\tau = \frac{V\rho}{W_i} \quad (2-23)$$

The dynamic models used for the reactor core and steam generators are described in sections 2.5 and 2.16, respectively.

#### Nomenclature

$T_i$	Temperature of fluid at inlet to plenum or section of piping, F
$T_o$	Temperature of fluid at outlet of plenum or section of piping, F
$V$	Volume of plenum or section of piping, ft <sup>3</sup>
$W_i$	Mass flow rate at inlet of plenum or section of piping, lbm/s
$W_o$	Mass flow rate at outlet of plenum, lbm/s
$\rho$	Density for each plenum or section of piping, input as a constant, lbm/ft <sup>3</sup>
$\tau$	Delay time, s

#### 2.8. Average Temperature Calculation for ICS

The integrated control system (ICS) requires average temperature for the primary system. This is computed from the hot and cold leg temperatures at the inlet and outlet of the steam generators. The control system selects either loop, and the measured average temperature is then calculated from the following differential equation which approximates the temperature measuring device characteristic:

$$\frac{dT_{avgm}}{dt} = \frac{1}{\tau_{ts}} (T_m - T_{avgm}), \quad (2-24)$$

$$T_m = \frac{1}{2} (T_{ha} + T_{ca}) \quad \text{if loop A is selected,} \quad (2-25)$$

$$T_m = \frac{1}{2} (T_{hb} + T_{cb}) \quad \text{if loop B is selected.} \quad (2-26)$$

#### Nomenclature

$T_{avgm}$	Measured value of primary average temperature, F
$T_{ca}$	Temperature of loop A cold leg, F
$T_{cb}$	Temperature of loop B cold leg, F
$T_{ha}$	Temperature of loop A hot leg, F
$T_{hb}$	Temperature of loop B hot leg, F

$T_m$  Actual value of primary average temperature, F  
 $\tau_{ts}$  Time constant of temperature measurement system, F

## 2.9. Primary System Pressure

A single pressure is used in the primary system model and is defined as the pressure that exists at the surge line connection to the loop A hot leg. The change in system pressure is calculated by considering the effects of thermal expansion of the primary fluid and net mass addition to the primary system, excluding the pressurizer. The cumulative effect of thermal expansion is considered for all subvolumes of the primary system.

$$\frac{dP_{rc}}{dt} = \frac{W_{mu} - W_{ld} - W_{sp} - W_{su} - \sum_i V_i \left( \frac{\partial \rho}{\partial T} \right)_P \frac{dT_i}{dt}}{V_{rc} \left( \frac{\partial \rho}{\partial P} \right)_T} \quad (2-27)$$

## Nomenclature

$P_{rc}$  Primary system pressure measured at pressure tap at surge line inlet, psia  
 $T_i$  Arithmetic mean of temperature of fluid at the inlet and outlet of subvolume i, F  
 $V_i$  Volume of subvolume i of the primary system, ft<sup>3</sup>  
 $V_{rc}$  Total volume of the primary system excluding the pressurizer, ft<sup>3</sup>  
 $W_{ld}$  Letdown flow rate out of the primary system, lbm/s  
 $W_{mu}$  Makeup flow rate into the primary system, lbm/s  
 $W_{sp}$  Spray flow rate into pressurizer, lbm/s  
 $W_{su}$  Surge flow rate into pressurizer, lbm/s  
 $(\partial \rho / \partial T)_P$  Function of temperature  $T_i$  defined in POWER TRAIN coding, lbm/ft<sup>3</sup>-°F  
 $(\partial \rho / \partial P)_T$  Function of pressure  $P_{rc}$  defined in POWER TRAIN coding, lbm/ft<sup>3</sup>-psi

## 2.10. Surge Line Flow

The surge line flow model assumes that the pressure drop across the surge line is due to elevation, friction, and shock losses. The following equation is solved for  $W_{su}$ :

$$P_{rc} = P_p + \Delta P_{esl} + K_{fsl} |W_{su}| W_{su}. \quad (2-28)$$

### Nomenclature

$K_{fsl}$	Equivalent dynamic $\Delta P$ coefficient in surge line, $\text{psi}/(\text{lbm/s})^2$
$\Delta P_{esl}$	Elevation $\Delta P$ for height difference between surge line connection and pressurizer, $\text{psi}$
$P_p$	Pressurizer pressure, $\text{psia}$
$P_{rc}$	Primary system pressure measured at pressure tap at surge line inlet, $\text{psia}$
$W_{su}$	Surge flow rate into pressurizer, $\text{lbm/s}$

### 2.11. Pressurizer

The pressurizer is modeled with a lower liquid region, an upper liquid region, and a vapor region. Each region may have a different temperature, but all are characterized by the same pressure. The pressure is determined by the density and enthalpy of the vapor region.

The pressurizer model includes the effects of sprays, heaters, surge flows, pilot valves, relief valves, and the makeup system, whose flow is regulated by the calculated liquid level in the pressurizer.

The mass and energy transfer effects modeled include

- Mixing of upper and lower liquid regions
- Evaporation and boiling of upper region liquid to vapor
- Condensation of vapor on upper liquid region surface
- Condensation of vapor on walls
- Condensation of vapor on spray

All these effects are modeled as surface phenomena.

#### 2.11.1. Mass Balance

During steady-state operation there is only one liquid region in the pressurizer. This liquid makes up what is called the upper region. It is assumed that during an insurge the incoming fluid will not mix instantly with the fluid already in the pressurizer. A new region, called the lower region, is created at this time. The mass of the lower region will decay exponentially as it mixes with the liquid in the upper region. Therefore, for a single insurge,

two mass balances evolve that will decay into a single region as steady state is approached. An outsurge will drain the lower region before the upper region mass is affected.

Whenever there is an insurge or whenever the mass of the lower liquid region has not yet decayed to zero, the mass balance for the lower liquid region is given by

$$\frac{dM_{pw1}}{dt} = W_{su} - \lambda_{pw1} M_{pw1}. \quad (2-29)$$

Mass flow from sprays, condensation on the wall, and condensation on the sprays are all assumed to go directly to the upper liquid region. Therefore, the mass balance for the upper region is given by

$$\frac{dM_{pw2}}{dt} = \lambda_{pw1} M_{pw1} + W_{su} B_{139} \overline{B_{140}} + W_{c4} + W_{c5} + W_{c6} - W_b + W_{sp}. \quad (2-30)$$

The mass balance for the vapor region is given by

$$\frac{dM_{ps}}{dt} = W_b - W_{c4} - W_{c5} - W_{c6} - W_r - W_{pr}. \quad (2-31)$$

### 2.11.2. Energy Balance

The conservation of energy equation can be written for each of the three regions in the pressurizer. It is assumed that there is no heat transfer with walls or across interfaces in the lower liquid region. For an insurge the flow enters at the hot leg enthalpy, and for an outsurge the flow exits at the bulk average enthalpy of the lower liquid region. When fluid from the lower liquid region mixes with that from the upper liquid region, its enthalpy is assumed to be equal to the bulk average enthalpy of the lower liquid region.

$$\begin{aligned} \frac{dh_{pw1}}{dt} = & \left[ (h_{HL} B_{138} + h_{pw1} B_{139}) W_{su} - M_{pw1} \lambda_{pw1} h_{pw1} \right. \\ & \left. - h_{pw1} \frac{d}{dt} M_{pw1} \right] \frac{B_{140}}{M_{pw1}}. \end{aligned} \quad (2-32)$$

Heat is added by the pressurizer heaters in the upper liquid region. Liquid is boiled off at the specific enthalpy of saturated vapor. Spray flow is assumed to be heated to saturation by the vapor and is added directly to the upper liquid region along with the condensate on the spray, all at the specific

enthalpy of saturated liquid. Condensation on the vapor-liquid interface adds not only condensate but also the heat of vaporization to the upper liquid region. Condensation on the walls adds liquid to the upper liquid region at the specific enthalpy of saturated liquid. The heat of vaporization from condensation on the walls is assumed to be taken up by the wall and lost. An out-surge affects the upper liquid region only when there is no mass in the lower liquid region, in which case the outsurge occurs at the bulk average enthalpy of the upper liquid region. The insurge will be added to the lower liquid region and mix with the upper liquid region according to a mixing factor:

$$\begin{aligned} \frac{dh_{pw2}}{dt} = & \left[ Q_{htr} - W_b h_g + (W_{c4} + W_{sp}) h_f + W_{c5} h_s + W_{c6} h_f \right. \\ & \left. + W_{su} h_{pw2} \frac{B_{139}}{B_{140}} + \lambda_{pw1} \frac{M_{pw1}}{M_{pw2}} h_{pw1} - h_{pw2} \frac{dM_{pw2}}{dt} \right] / M_{pw2}. \end{aligned} \quad (2-33)$$

Specific enthalpy in the vapor region is increased when there is boiling of the upper liquid region and is decreased through condensation on walls, sprays, and the surface of the upper liquid region. Energy is lost when mass is vented through the pilot and relief valves.

$$\frac{dh_s}{dt} = [W_b h_g - (W_{c4} + W_r + W_{pr} + W_{c5} + W_{c6} + \dot{M}_{ps}) h_s + V_{ps} \dot{P}] / M_{ps}. \quad (2-34)$$

By solving the conservation of mass for the vapor region mass,  $W_b$ , and substituting the result in the energy equation, a substantially simplified energy equation is obtained:

$$\frac{dh_s}{dt} = \frac{W_b (h_g - h_s)}{M_{ps}} + \frac{V_{ps}}{M_{ps}} \dot{P}. \quad (2-35)$$

### 2.11.3. Mass and Energy Transfer Processes

The pressurizer pressure is a function of the local density and enthalpy for a positive level in the pressurizer. Should the pressurizer empty, the pressure becomes a function of only the hot leg temperature.

Upper liquid region boiling occurs when the liquid enthalpy exceeds the saturation enthalpy corresponding to the pressurizer pressure. It is assumed that the boiling rate is a linear function of the difference between the local and saturation enthalpies.

$$W_b = K_{2,6} (h_{pw2} - h_f) B_{142} \quad (2-36)$$

The condensation rate of vapor on the spray depends on the saturation enthalpy, the spray inlet enthalpy (which is the enthalpy of the cold leg), and the spray flow rate. It is assumed that all spray flow is heated to the saturation temperature by condensation of vapor on the spray. This assumption implies that the condensation rate is such that the heat of vaporization given up is sufficient to raise the enthalpy of the spray from the inlet value to the saturation value.

$$W_{cs} = \frac{W_{sp} (h_f - h_{CL})}{h_s - h_f} \quad (2-37)$$

The condensation rate of vapor on the liquid surface and on the walls is limited by the heat of vaporization removal rate. It is assumed that this heat is removed in proportion to the  $\Delta T$  between the surface and the bulk vapor temperature. The liquid surface temperature is assumed to be equal to the bulk liquid temperature of the upper liquid region. The wall temperature and the heat transfer factors (UA) are constants included in the input data.

$$W_{cs} = \overline{B}_{142} (T_{ps} - T_{pw2}) [(UA)_{ps}] / (h_s - h_f) \quad (2-38)$$

$$W_{cw} = \overline{B}_{143} (T_{ps} - T_{pm2}) [(UA)_{pm2}] / (h_s - h_f) \quad (2-39)$$

#### 2.11.4. State Relations and Water Properties

To determine the state of each pressurizer region, the following additional relations are needed together with functions for water properties.

The volume of the liquid region is the sum of the volume of the two lower regions:

$$V_{pw} = \frac{M_{pw1}}{\rho_{pw1}} + \frac{M_{pw2}}{\rho_{pw2}} \quad (2-40)$$

The volume of the steam region may then be found by

$$V_{ps} = K_{2,5} - V_{pw} \quad (2-41)$$

The density of the steam region is

$$\rho_{ps} = M_{ps} / V_{ps} \quad (2-42)$$

The water properties needed by the pressurizer model are expressed as polynomial fits to the ASME Steam Tables.

$$T_{\text{sat}} = f_{201}(h_f) \quad (2-43)$$

$$h_f = f_{202}(P) \quad (2-44)$$

$$h_g = f_{203}(P) \quad (2-45)$$

$$\rho_f = f_{204}(P) \quad (2-46)$$

$$h_f = f_{207}(T) \quad (2-47)$$

$$T = f_{208}(v, h) \quad (2-48)$$

$$P = f_{209}(v, h) \quad (2-49)$$

#### Nomenclature

Symbols beginning with B are logical variables, which have a value of either 1 or 0 to correspond with true or false. In equations, a vinculum is used over logical variables to represent the logical inverse.

$$B_{138} = 1 \text{ when } W_{su} > 0, \\ = 0 \text{ otherwise}$$

$$B_{139} = 1 \text{ when } W_{su} \leq 0, \\ = 0 \text{ otherwise}$$

$$B_{140} = 1 \text{ when } M_{pwi} > 0, \\ = 0 \text{ otherwise}$$

$$B_{142} = 1 \text{ when } h_{pw2} > h_f, \\ = 0 \text{ otherwise}$$

$$B_{143} = 1 \text{ when } T_{ps} > T_{pm2}, \\ = 0 \text{ otherwise}$$

e Specific internal energy, Btu/lbm

h Specific enthalpy, Btu/lbm

$h_{CL}$  Specific enthalpy of water in cold leg, Btu/lbm

$h_f$  Specific enthalpy of liquid at saturation, Btu/lbm

$h_g$  Specific enthalpy of vapor at saturation, Btu/lbm

$h_{HL}$  Specific enthalpy of water in hot leg, Btu/lbm

$h_{pw1}$	Specific enthalpy of water in lower liquid region of pressurizer, Btu/lbm
$h_{pw2}$	Specific enthalpy of water in upper liquid region of pressurizer, Btu/lbm
$h_s$	Specific enthalpy of vapor in the steam region of pressurizer, Btu/lbm
$K_{2.5}$	Total pressurizer volume, $ft^3$
$K_{2.6}$	Equivalent coefficient for boil-off in pressurizer, $lb/s-^{\circ}F$
$M$	Total mass in a control volume, lbm
$M_{ps}$	Total mass in steam region of pressurizer, lbm
$M_{pw1}$	Total mass in lower liquid region of pressurizer, lbm
$M_{pw2}$	Total mass in upper liquid region of pressurizer, lbm
$p$	Pressure, psia
$Q_{htr}$	Heater power, Btu/s
$T_{ps}$	Temperature of steam region of pressurizer, F
$T_{pw1}$	Temperature of lower liquid region of pressurizer, F
$T_{pw2}$	Temperature of upper liquid region of pressurizer, F
$(UA)_{ps}$	Steam-to-liquid condensation heat transfer rate, $Btu/s-^{\circ}F$
$(UA)_{pm2}$	Steam-to-wall condensation heat transfer rate, $Btu/s-^{\circ}F$
$v$	Specific volume, $ft^3/lbm$
$V_{ps}$	Volume of steam region in the pressurizer, $ft^3$
$V_{pw}$	Volume of both liquid regions in the pressurizer, $ft^3$
$W_b$	Rate of boil-off from upper liquid region to steam region of pressurizer, lbm/s
$W_{c4}$	Rate of condensation of vapor onto spray droplets in pressurizer, lbm/s
$W_{c5}$	Rate of condensation of steam on liquid surface in pressurizer, lbm/s
$W_{c6}$	Rate of condensation of steam on pressurizer wall, lbm/s
$W_{pr}$	Flow rate through power-operated relief valve, lbm/s

$W_r$	Flow rate through Code Safety valve, lbm/s
$W_{sp}$	Spray flow rate, lbm/s
$W_{su}$	Surge flow rate, lbm/s
$\lambda_{pw1}$	Mixing factor for lower and upper liquid regions of the pressurizer, $s^{-1}$
$\rho_{ps}$	Density of steam in the steam region of pressurizer, lbm/ft <sup>3</sup>

### 2.12. Pressurizer Heaters

Four banks of heaters are modeled; their total energy output is the sum of the contributions from each one. The first bank is operated proportionally, the remaining three are on-off controlled with different cut-on and cut-off temperatures and power capabilities. A 30-second time constant is associated with each heater to account for heat storage in the heater. In addition, all heaters are cut off should pressurizer level fall below the level specified by the input data.

### 2.13. Pressurizer Sprays

The spray valve opens and closes at a fixed velocity. The opening and closing pressure setpoints and the velocity are constants specified by the input data. Spray flow is given by the following equation, where  $f(N_{rcp})$  is a function of the number of reactor coolant pumps in operation:

$$W_{sp} = K_{2,33} X_{sp} f(N_{rcp})/100 \quad (2-50)$$

$$\frac{dX_{sp}}{dt} = \begin{cases} K_{2,42} & \text{if } P_p \geq K_{2,34} \\ 0 & \text{if } K_{2,35} < P_p < K_{2,34} \\ -K_{2,42} & \text{if } P_p \leq K_{2,35} \end{cases} \quad (2-51)$$

### Nomenclature

$f(N_{rcp})$	Function of number of reactor coolant pumps in operation, dimensionless
$K_{2,33}$	Maximum spray flow rate, lbm/s
$K_{2,34}$	Spray cut-on pressure, psia
$K_{2,35}$	Spray cut-off pressure, psia

$K_{2.42}$	Pressurizer spray valve velocity, % stroke/s
$P_p$	Pressurizer pressure, psia
$W_{sp}$	Spray flow rate, lbm/s
$X_{sp}$	Spray valve position limited to value between 0 and 100%, % stroke

#### 2.14. Pressurizer Code Safety and Power-Operated Relief Valves

The Code Safety valve opens and closes immediately upon actuation. The power-operated relief valve ramps open and closed at a rate of 200% per second. The opening and closing pressures and the flow capacity of each valve are specified by the input data.

#### 2.15. Makeup System

A level is calculated in the pressurizer according to the following equation:

$$L_p = K_{2.3} V_{pw} + K_{2.4} \quad (2-52)$$

A proportional-integral controller monitors this pressurizer level and controls the makeup flow demand. The flow follows the demand with a first-order lag response. Flow is constrained to be less than an input constant,  $K_{2.31}$ .

$$\frac{dW_{mu}}{dt} = \frac{(W_{mu})_D - W_{mu}}{\tau_{mu}} \quad (2-53)$$

$$(W_{mu})_D = K_{2.27} [K_{2.28} (K_{2.29} - L_p) + K_{2.30} \int (K_{2.29} - L_p) dt] \quad (2-54)$$

#### Nomenclature

$K_{2.3}$	Level to volume equivalent in cylindrical section of pressurizer, in./ft <sup>3</sup>
$K_{2.4}$	Level equivalent of pressurizer volume below lower level sensing tap, in.
$K_{2.27}$	Conversion factor, level error to makeup flow demand, lbm/s-in.
$K_{2.28}$	Proportional gain, level error, dimensionless
$K_{2.29}$	Pressurizer level setpoint, in.
$K_{2.30}$	Integral gain, level error, s <sup>-1</sup>

$K_{2,31}$	Maximum makeup flow rate, lbm/s
$L_p$	Pressurizer level, in.
$V_{pw}$	Volume of both liquid regions in the pressurizer, ft <sup>3</sup>
$W_{mu}$	Makeup flow rate, lbm/s
$(W_{mu})_D$	Makeup flow demand, lbm/s
$\tau_{mu}$	Makeup flow time constant, s

## 2.16. Steam Generator

Two steam generators provide the interface between the primary and secondary flow loops of the power plant by transferring heat from the primary to the secondary side. Primary water flows from the reactor into the upper plenum of the steam generators and is cooled as it flows through the steam generator tubes to the lower plenum and is pumped back to the reactor. Secondary fluid is pumped from the feedwater train into the bottom of the tube bundle and is successively heated, boiled, and superheated as it flows around the tubes until it exits the top of the tube bundle through an annulus into the steam lines.

POWER TRAIN provides the option to model either an OTSG or an IEOTSG. The term integral economizer refers to the fact that the feedwater enters the tube bundle in a subcooled state and is heated to saturation in the economizer section of the steam generator (typically, the first 8 feet of the tube bundle). The designation OTSG is used here to refer to a once-through steam generator in which feedwater is preheated to saturation by recirculated steam in the downcomer before being introduced into the tube bundle.

The model for the steam generators is described by nonlinear partial differential equations (PDE) involving both time and one spatial dimension. The boundary conditions required by the model are as follows:

1. Primary flow rate.
2. Temperature of fluid at primary inlet.
3. Secondary inlet flow rate.
4. Temperature of fluid at secondary inlet.
5. Pressure at secondary outlet.

The following modeling assumptions are made:

1. The flow areas on both primary and secondary sides are constant.
2. The primary fluid can be considered to be incompressible; hence, there is no flow variation in the channel.
3. Axial heat conduction in the tube metal is negligible when compared to that in the radial direction.
4. The secondary fluid state may be evaluated as a function of local enthalpy at an average pressure during a time step.
5. The overall dynamic behavior of the IEOTSG can be represented by an average channel.

A complete solution of the equations modeling steam generator dynamics consists of calculating the following variables as a function of time and the one spatial dimension modeled:

1. Primary side flow rate.
2. State of fluid on primary side.
3. Temperature of tube metal.
4. Secondary side flow rate.
5. State of fluid on secondary side.

The primary and secondary energy balances are coupled via the tube metal temperature distribution. The primary fluid temperature is determined as an initial value problem from flow inlet to outlet for each time step. Then the secondary fluid temperature is calculated and the metal temperature is updated. The steam generator equations are solved using the continuous space discrete time (CSDT) method, where finite differencing is used for the time derivatives and continuous integration is used for the spatial derivatives.

#### 2.16.1. Primary Side Model

By assumption 2 above, the flow rate on the primary side of the steam generator is constant in space. This flow rate, which is calculated by the primary system flow model (see section 3.6), may be in either the normal or reverse direction. The steam generator model can accommodate either case.

The primary side fluid is assumed to always be subcooled single-phase liquid. In general, the state of the fluid can be specified by specifying its pressure and temperature. Therefore, the conservation of energy equation for the primary fluid gives us the following PDE for the primary temperature:

$$\frac{\partial T_p}{\partial t} + \frac{W_p}{\rho_p A_p} \frac{\partial T_p}{\partial x} = \frac{p_p U_{pm}}{\rho_p A_p C_p} (T_m - T_p). \quad (2-55)$$

### 2.16.2. Tube Metal Model

The effect of metal conductivity is included in the values of  $U_{pm}$  and  $U_{sm}$ . By assumption 3 above, we neglect axial heat conduction along the tubes and write the following energy equation for the tube metal:

$$\frac{\partial T_m}{\partial t} = \frac{p_p U_{pm}}{\rho_m C_m A_m} (T_p - T_m) + \frac{p_s U_{sm}}{\rho_m C_m A_m} (T_s - T_m). \quad (2-56)$$

### 2.16.3. Secondary Side Model

The modeling of the secondary side is complicated by the fact that the fluid changes state from subcooled liquid to a boiling mixture and finally to superheated steam as it flows from the bottom to the top of the tube bundle. By assumption 5 we will assume (for the purposes of evaluating state properties) that the pressure,  $P_{sg}$ , is the same for the entire length of the tube bundle and is equal to the pressure at the secondary outlet:

$$P_{sg} = P_{sgo}. \quad (2-57)$$

$P_{sgo}$  is calculated by the steam line model (see section 2.18) and is provided as a boundary condition to the steam generator model. This assumption of constant pressure along the length of the channel implies that the state of the fluid is specified when enthalpy along the channel is specified.

With respect to calculating flow and density, the secondary side of the steam generator can be divided into three regions according to the state of the secondary fluid: the subcooled region, the boiling region, and the superheated region. The boundaries of these regions may change from one time step to the next. The boundary between the subcooled and boiling regions occurs at the height where

$$h_s = h_f(P_{sg}) \quad \text{at } z = z_{sc}. \quad (2-58)$$

The boundary between the boiling mixture and superheated vapor occurs at the height where

$$h_s = h_g(P_{sg}) \quad \text{at } z = z_b. \quad (2-59)$$

The pressure drop across the steam generator is calculated using a pressure balance. Kinetic energy and inertial terms are assumed to be negligible. The pressure at the inlet of each steam generator is calculated for use by the feedwater system model and is not used by any other equation in the steam generator model, i.e.,

$$P_{sgf} = P_{sgo} + \Delta P_g + \Delta P_f. \quad (2-60)$$

The gravity head is the spatial integral of density,

$$\Delta P_g = \frac{g}{144g_c} \int_0^L \rho_s dz. \quad (2-61)$$

The shock and friction losses associated with each region are modeled using a nominal pressure gradient for a reference value of flow. The pressure gradient is modeled as being proportional to the square of the actual flow. The integral of the pressure gradient is evaluated as follows:

$$\Delta P_f = \int_0^{z_{sc}} \left( \frac{\Delta P}{\Delta l} \right)_{sc} \frac{W_s |W_s|}{W_o^2} dz + \int_{z_{sc}}^{z_b} \left( \frac{\Delta P}{\Delta l} \right)_b \frac{W_s |W_s|}{W_o^2} dz + \int_{z_b}^L \left( \frac{\Delta P}{\Delta l} \right)_{sh} \frac{W_s |W_s|}{W_o^2} dz. \quad (2-62)$$

The flow on the secondary side is found by integrating the continuity equation, which is usually written in the following form:

$$\frac{1}{A_s} \frac{\partial}{\partial z} W_s + \frac{\partial}{\partial t} \rho_s = 0. \quad (2-63)$$

Since  $\rho_s = \rho_s(h_s, P_{sg})$ , an equivalent form of the continuity equation is

$$\frac{\partial}{\partial z} W_s = -A_s \left( \frac{\partial \rho_s}{\partial h_s} \right)_P \frac{\partial}{\partial t} h_s - A_s \left( \frac{\partial \rho_s}{\partial P_{sg}} \right)_h \frac{\partial}{\partial t} P_{sg}. \quad (2-64)$$

The derivatives of density, pressure, and enthalpy are given by the state relations, the boundary conditions, and the conservation of energy equation, respectively. With the assumption that  $\partial P_{sg} / \partial t$  is small in comparison with  $Q_s$ , the energy equation becomes

$$\frac{\partial}{\partial t} \rho_s h_s + \frac{1}{A_s} \frac{\partial}{\partial z} W_s h_s - Q_s = 0. \quad (2-65)$$

Combine equation 2-65 with the conservation of mass equation 2-63 to obtain the form of the energy equation used to calculate the enthalpy along the channel:

$$\rho_s \frac{\partial}{\partial t} h_s + \frac{W_s}{A_s} \frac{\partial}{\partial z} h_s = Q_s. \quad (2-66)$$

#### 2.16.4. Heat Transfer Relations

Heat transfer relations are needed by the energy equation for the primary side, the tube metal, and the secondary side of the steam generator. Half of the thermal resistivity of the tube metal is combined with the primary film heat transfer coefficient, and the other half is combined with the secondary film heat transfer coefficient.

The metal-to-primary heat transfer is modeled with a single constant overall heat transfer coefficient ( $U_{pm}$  provided as input data):

$$Q_p = \frac{P_p}{A_p} U_{pm} [T_m(z,t) - T_p(z,t)]. \quad (2-67)$$

The model for the metal-to-secondary heat transfer is more complex. The heat transfer equation is the same, but the form of the heat transfer coefficient is more complicated:

$$Q_s = \frac{P_s}{A_s} U_{sm} [T_m(z,t) - T_s(z,t)]. \quad (2-68)$$

Modeling the metal-to-secondary heat transfer requires division of the secondary side into three regions to represent three different heat transfer processes that occur in each region: subcooled, nucleate boiling, and superheat. The boundary between these regions will change from one time step to the next. The boundary between subcooled and nucleate boiling is at the height where

$$h_s = h_g(P_{sg}). \quad (2-69)$$

The boundary between nucleate boiling and superheat is at the height where

$$X_g \geq \text{DNBX}. \quad (2-70)$$

DNBX is an input constant representing that quality value above which film boiling is expected to occur.

For the subcooled region the heat transfer coefficient is a weighted sum of a forced convection term and the nucleate boiling coefficient.

$$U_{sm} = K_{sc} W_s(z,t) + 0.5 U_b. \quad (2-71)$$

The nucleate boiling coefficient is based on the model developed by Thom<sup>3</sup> and is expressed as

$$U_{sm} = U_b \quad (2-72)$$

$$U_b = K_1 \phi / (T_m - T_s) \quad (2-73)$$

$$\phi = \frac{k}{C_t} (T_m - T_{sat} - \Delta T_{sat}) \quad (2-74)$$

$$C_t = D_o \ln[2 D_o / (D_o + D_i)] / 2 \quad (2-75)$$

$$\Delta T_{sat} = \frac{4.32 \phi^{0.5}}{e^{P/1260}}. \quad (2-76)$$

The forced convection coefficient for the superheated region is expressed as

$$U_{sm} = K_{sh} W_s(z,t). \quad (2-77)$$

#### 2.16.5. State Relations

The following state relations are approximated by polynomials whose coefficients are built into the simulation:

$$h_f = f(P_{sg}) \quad (2-78)$$

$$h_g = f(P_{sg}) \quad (2-79)$$

$$T_{sc} = f(h_s) \quad (2-80)$$

$$T_{sh} = f(P_{sg}, h_s) \quad (2-81)$$

$$T_{sat} = f(P_{sg}) \quad (2-82)$$

$$\rho_f = f(P_{sg}) \quad (2-83)$$

$$\rho_g = f(P_{sg}) \quad (2-84)$$

$$\rho_{sc} = f(T_{sc}) \quad (2-85)$$

$$\rho_{sh} = f(T_{sc}) \quad (2-86)$$

The evaluation of density for the boiling region is based on the "drift flux" two-phase void fraction correlation proposed by N. Zuber.<sup>4</sup> The following equations are solved for  $\rho_s$  in the boiling region:

$$\rho_s = \rho_f - \alpha(\rho_f - \rho_g) \quad (2-87)$$

$$\alpha = \frac{X}{C_o \left( \frac{\rho_g}{\rho_f} + \frac{\rho_f - \rho_g}{\rho_f} X \right) + \frac{A_s \rho_g \bar{V}_{gj}}{W_f}} \quad (2-88)$$

$$X = \frac{h - h_f}{h_g - h_f} \quad (2-89)$$

$$C_o = 1.0 \quad (2-90)$$

$$\bar{V}_{gj} = \left( 1 - \frac{\Delta X}{1 - X_c} \right) 1.41 \left[ \frac{\sigma_{ggc} (\rho_f - \rho_g)}{\rho_f^2} \right]^{1/4} \quad (2-91)$$

$$\Delta X = \begin{cases} X - X_c & \text{for } X \geq X_c \\ 0 & \text{for } X < X_c \end{cases} \quad (2-92)$$

$$W_f = W_{fw} - A_s \frac{d}{dt} \int_0^{L_{sc}} (\rho_f + \rho_{sc}) dz \quad (2-93)$$

Derivatives of density with respect to pressure and enthalpy are required to solve the conservation of mass equation (2-64). In the subcooled liquid and superheated steam regions these derivatives are approximated as constants calculated from input data.

$$\left( \frac{\partial \rho_s}{\partial P_{sg}} \right)_{sc} = 0 \quad (2-94)$$

$$\left( \frac{\partial \rho_s}{\partial P_{sg}} \right)_{sh} = DRHODP \quad (2-95)$$

$$\left( \frac{\partial \rho_s}{\partial h_s} \right)_{sc} = \frac{R1 - R2}{H1 - F2} \quad (2-96)$$

$$\left(\frac{\partial \rho_s}{\partial h_s}\right)_{sh} = \frac{R4 - R5}{G2 - H4} \quad (2-97)$$

### Nomenclature

DRHODP	$\partial \rho / \partial P$ at enthalpy of 1234.7, lbm/ft <sup>3</sup> -psi
R1	Subcooled density at FW1 and P2, lbm/ft <sup>3</sup>
R2	Saturated fluid density at T2 and P2, lbm/ft <sup>3</sup>
F2	Saturated fluid enthalpy at P2, Btu/lbm
H1	Subcooled enthalpy at FW1 and P2, Btu/lbm
R4	Saturated gas density at T2 and P2, lbm/ft <sup>3</sup>
R5	Superheated density at T4 and P2, lbm/ft <sup>3</sup>
H4	Superheated enthalpy at T4 and P2, Btu/lbm
G2	Saturated gas enthalpy at P2, Btu/lbm
FW1	Minimum feedwater temperature, F
P2	Expected normal pressure, psia
T2	Saturation temperature at P2, psia
T4	Expected full power steam temperature, F

The calculation of the density derivatives is more involved in the boiling region. The expressions for each derivative can be derived by differentiating the expression for density given by equations 2-87 through 2-93. The  $\partial \rho / \partial h$  for the boiling region is calculated by the following set of equations:

$$\left(\frac{\partial \rho}{\partial h}\right)_P = -(\rho_f - \rho_g) \left(\frac{\partial \alpha}{\partial h}\right)_P \quad (2-98)$$

$$\left(\frac{\partial \alpha}{\partial h}\right)_P = \frac{1}{V^2} \left[ V - C_o X \left( \frac{\rho_f - \rho_g}{\rho_f} \right) \right] \left(\frac{\partial X}{\partial h}\right)_P \quad (2-99)$$

$$\left(\frac{\partial X}{\partial h}\right)_P = \frac{1}{h_g - h_f} \quad (2-100)$$

$$V = C_o \left( \frac{\rho_g}{\rho_f} + \frac{\rho_f - \rho_g}{\rho_f} X \right) + \frac{A_s \rho_g \bar{V}_g}{W_f} \quad (2-101)$$

The  $\partial\rho/\partial P$  for the boiling region is calculated by the following set of equations:

$$\left(\frac{\partial\rho}{\partial P}\right)_h = \frac{d\rho_f}{dP} - \alpha \frac{d}{dP} (\rho_f - \rho_g) - (\rho_f - \rho_g) \left(\frac{\partial\alpha}{\partial P}\right)_h \quad (2-102)$$

$$\left(\frac{\partial\alpha}{\partial P}\right)_h = \frac{1}{v^2} \left[ \left(\frac{\partial X}{\partial P}\right)_h v - X \left(\frac{\partial v}{\partial P}\right)_h \right] \quad (2-103)$$

$$\left(\frac{\partial X}{\partial P}\right)_h = - \left( \frac{1}{h_g - h_f} \right) \left[ X \frac{dh_g}{dP} + (1 - X) \frac{dh_f}{dP} \right] \quad (2-104)$$

$$\left(\frac{\partial v}{\partial P}\right)_h = C_o \left[ \frac{d}{dP} \left( \frac{\rho_g}{\rho_f} \right) + \frac{\rho_f - \rho_g}{\rho_f} \left(\frac{\partial X}{\partial P}\right)_h + X \frac{d}{dP} \frac{\rho_f - \rho_g}{\rho_f} \right] + \frac{\rho_g A_s}{W_f} \left( \frac{\partial \bar{v}_g}{\partial P} \right)_h \quad (2-105)$$

### Nomenclature

A	Cross-sectional area, ft <sup>2</sup>
C	Specific heat, Btu/lbm-°F
C <sub>o</sub>	Distribution parameter, dimensionless
C <sub>t</sub>	Equivalent diameter, ft
D <sub>i</sub>	Tube inside diameter, ft
D <sub>o</sub>	Tube outside diameter, ft
DNBX	Quality at which DNB occurs, dimensionless
g	Gravitational acceleration, ft/s <sup>2</sup>
g <sub>c</sub>	Gravitational conversion constant, lbm-ft/lbf-s <sup>2</sup>
h	Specific enthalpy, Btu/lbm
h <sub>f</sub>	Specific enthalpy of saturated liquid, Btu/lbm
h <sub>g</sub>	Specific enthalpy of saturated vapor, Btu/lbm
k	Thermal conductivity of tube metal, Btu/s-ft-°F
K <sub>sc</sub>	Subcooled region forced convection heat transfer factor, Btu/lbm-°F
K <sub>sh</sub>	Superheated region forced convection heat transfer factor, Btu/lbm-°F
K <sub>1</sub>	Multiplier on Thom's correlation, dimensionless
L	Length of steam generator tube bundle, ft

$P$	Perimeter, ft
$P$	Pressure, psia
$\Delta P_g$	Gravity head, psi
$\Delta P_f$	Shock and friction pressure drop, psi
$P_{sg}$	Steam generator pressure for state property evaluation, psia
$P_{sgi}$	Steam generator pressure at feedwater inlet of steam generator, psia
$P_{sgo}$	Steam generator pressure at steam outlet of steam generator, psia
$\Delta P/\Delta L$	Unrecoverable pressure losses per unit length, psi/ft
$Q_p$	Rate of heat addition to primary side of steam generator, Btu/s
$Q_s$	Rate of heat addition to secondary side of steam generator, Btu/s
$T$	Temperature, F
$T_{sat}$	Secondary fluid saturation temperature, F
$\Delta T_{sat}$	Temperature differential, tube wall to secondary fluid, F
$U_b$	Metal-to-secondary heat transfer coefficient in boiling region, Btu/s-ft <sup>2</sup> -°F
$U_{pm}$	Primary-to-metal heat transfer coefficient, Btu/s-ft <sup>2</sup> -°F
$U_{sm}$	Secondary-to-metal heat transfer coefficient, Btu/s-ft <sup>2</sup> -°F
$V$	Intermediate term in drift-flux model, dimensionless
$\bar{V}_{gi}$	Drift velocity of vapor, ft/s
$W$	Mass flow rate, lbm/s
$W_f$	Equivalent flow rate from subcooled to boiling region, lbm/s
$W_o$	Reference mass flow rate for secondary, lbm/s
$x$	Distance measured from primary fluid inlet, ft
$\Delta X$	Difference between $X_s$ and $X_c$ , dimensionless
$X_c$	Threshold quality defined to smooth transition from boiling region to superheated region, dimensionless
$z$	Distance measured from bottom of tube bundle, ft
$z_{sc}$	Height above bottom tubesheet of interface between subcooled and boiling region, ft

$z_b$	Height above bottom tubesheet of interface between boiling and superheat region, ft
$\alpha$	Void fraction, dimensionless
$\rho$	Density, lbm/ft <sup>3</sup>
$\rho_f$	Density of saturated liquid, lbm/ft <sup>3</sup>
$\rho_g$	Density of saturated vapor, lbm/ft <sup>3</sup>
$\sigma$	Surface tension, lbf/ft
$\phi$	Heat flux, Btu/s-ft <sup>2</sup>

#### Subscripts

b	Boiling region
m	Tube metal
p	Primary side
s	Secondary side
sc	Subcooled region
sh	Superheated region

#### 2.17. Downcomer Model

An option is available in POWER TRAIN to simulate the steam generator with a preheating feedwater chamber. This downcomer preheats the feedwater to saturated conditions (using recirculated superheated steam) before it enters the tube bundle. Therefore, when using this option, the subcooled region is eliminated from the simulation and the following model is substituted. The following assumptions are made in deriving this model:

1. The aspiration steam maintains the downcomer at saturated conditions.
2. There is a distinct saturated fluid level.
3. Spatial momentum and transient fluid acceleration pressure differentials are negligible.
4. The time rate of change of the saturated liquid and vapor properties (specifically density and enthalpy) are negligible.

The equations that describe the dynamics of the downcomer follow.

### 2.17.1. Conservation of Mass

$$A_{dc} \frac{d}{dt} [\rho_g (L_{dc} - L_\ell) + \rho_f L_\ell] = W_{fw} + W_{asp} - W_f \quad (2-106)$$

where

- $A_{dc}$  = downcomer area, constant,  $\text{ft}^2$ ,
- $L_{dc}$  = downcomer height, constant,  $\text{ft}$ ,
- $L_\ell$  = saturated fluid level,  $\text{ft}$ ,
- $\rho_g$  = saturated vapor density,  $\text{lbm}/\text{ft}^3$ ,
- $\rho_f$  = saturated fluid density,  $\text{lbm}/\text{ft}^3$ ,
- $W_{fw}$  = feedwater flow rate entering downcomer,  $\text{lbm}/\text{s}$ ,
- $W_{asp}$  = vapor aspiration flow rate,  $\text{lbm}/\text{s}$ ,
- $W_f$  = saturated fluid flow rate leaving downcomer,  $\text{lbm}/\text{s}$ .

The continuity equation is now solved for the time rate of change of the fluid level by utilizing assumption 4 above:

$$\frac{d}{dt} (L_\ell) = \frac{1}{A_{dc}} \left[ \frac{W_{fw} + W_{asp} - W_f}{\rho_f - \rho_g} \right] \quad (2-107)$$

### 2.17.2. Conservation of Energy

$$A_{dc} \frac{d}{dt} [\rho_g h_g (L_{dc} - L_\ell) + \rho_f h_f L_\ell] = W_{fw} h_{fw} + W_{asp} h_{asp} - W_f h_f \quad (2-108)$$

where (in addition to previously defined quantities)

- $h_g$  = saturated vapor enthalpy,  $\text{Btu}/\text{lbm}$ ,
- $h_f$  = saturated fluid enthalpy,  $\text{Btu}/\text{lbm}$ ,
- $h_{fw}$  = feedwater enthalpy,  $\text{Btu}/\text{lbm}$ ,
- $h_{asp}$  = aspiration flow enthalpy,  $\text{Btu}/\text{lbm}$ .

This equation is now solved for the time rate of change of the fluid level by once again using assumption 4 above:

$$\frac{d}{dt} (L_\ell) = \frac{1}{A_{dc}} \left[ \frac{W_{fw} h_{fw} + W_{asp} h_{asp} - W_f h_f}{\rho_f h_f - \rho_g h_g} \right] \quad (2-109)$$

Now the aspiration flow rate required to keep the downcomer at saturated conditions can be found by equating 2-107 and 2-109 and solving for  $W_{asp}$ :

$$W_{asp} = \frac{W_f (h_f - K_1) - W_{fw} (h_{fw} - K_1)}{h_{asp} - K_1} \quad (2-110)$$

where

$$K_1 = \frac{\rho_f h_f - \rho_g h_g}{\rho_f - \rho_g} \quad (2-111)$$

### 2.17.3. Conservation of Momentum

The flow rate of saturated fluid ( $W_f$ ) which leaves the downcomer and enters the tube bundle is found by considering the pressure differentials that exist in the tube region and downcomer. Since the downcomer is open to the tube region at the bottom (inlet orifice) and at the top (aspiration port), the sum of the following pressure differentials must be zero:

$$\Delta P_{dcg} + \Delta P_{fwn} - \Delta P_{orf} - \Delta P_{otsg} - \Delta P_{asp} = 0 \quad (2-112)$$

where

$\Delta P_{dcg}$  = pressure differential due to fluid and vapor mass in downcomer, psi,

$\Delta P_{fwn}$  = pressure differential due to feedwater momentum force from inlet nozzle, psi,

$\Delta P_{orf}$  = pressure differential due to orifice flow rate, psi,

$\Delta P_{otsg}$  = pressure differential due to flow rate and mass in tube region, psi,

$\Delta P_{asp}$  = pressure differential due to aspiration flow rate, psi.

Neglecting spatial momentum and transient fluid acceleration effects (assumption 3),

$$\Delta P_{dcg} = \frac{g}{144g_c} [\rho_f L_l + \rho_g (L_{dc} - L_l)], \quad (2-113)$$

$$\Delta P_{orf} = R_{orf} \left( \frac{|W_f| W_f}{\rho_f} \right), \quad (2-114)$$

$$\Delta P_{asp} = R_{asp} \left( \frac{|W_{asp}| W_{asp}}{\rho_g} \right), \quad (2-115)$$

$$\Delta P_{fwn} = R_{fdc} \left( \frac{W_{fw}^2}{\rho_{fw}} \right). \quad (2-116)$$

The tube region pressure drop ( $\Delta P_{otsg}$ ) is discussed in section 2.16.3 (equations 2-61 and 2-62). It is calculated from the lower tubesheet to the aspiration port and input to the downcomer model;  $R_{orf}$ ,  $R_{asp}$ , and  $R_{fdc}$  above are equivalent friction, shock, and momentum coefficients determined from steady-state considerations.

Combining equations 2-112 and 2-114 and solving for the orifice flow rate (and requiring that  $W_f \geq 0$ ),

$$W_f = \left[ \frac{\rho_f}{R_{orf}} (\Delta P_{dcg} + \Delta P_{fwn} - \Delta P_{otsg} - \Delta P_{asp}) \right]^{1/2} \quad (2-117)$$

The equations above are implemented on the digital portion of the hybrid computer. The time rate of change of fluid level is calculated using Euler integration. The remaining equations are algebraic.

### 2.18. Steam Lines

Steam from the two steam generators flows into the steam lines and is combined in the turbine header before flowing onward through the turbine control valves to the turbine. Only one steam line is modeled per steam generator. The partial differential equations describing the flow of steam are solved using the discrete space continuous time method. The nodalization scheme is shown in Figure 2-1.

The steam line model requires that the following boundary conditions be provided:

1. Flow out of each steam generator.
2. Positions of all dump, bypass, and control valves.

The following assumptions are made in the model:

1. No heat is lost or gained through the walls.
2. The steam is always superheated.
3. The steam enthalpy is constant.

The steam line model consists of the solutions of equations to calculate the following variables as a function of time:

1. Pressure in every node.
2. Flows into and out of every node.

Assumptions 1 and 3 eliminate the need for an energy equation. Assumptions 2 and 3 allow the equation of state to be approximated as

$$\rho(x,t) = C_1 + C_2 P(x,t) \quad (2-118)$$

where  $C_1$  and  $C_2$  are constants derived from input data.

The conservation of mass equation takes the following form when this equation of state is used:

$$\frac{\partial}{\partial t} \rho + \frac{1}{A} \frac{\partial}{\partial x} W = 0. \quad (2-119)$$

Since steam enthalpy is assumed constant,  $\rho = f(P)$ , and we may write

$$\frac{\partial P}{\partial t} \frac{d\rho}{dP} + \frac{1}{A} \frac{\partial}{\partial x} W = 0. \quad (2-120)$$

The conservation of mass equations for all nodes are as follows (see Figure 2-1):

$$\frac{d}{dt} P_{sgoa} = \left( \frac{1}{ALC_2} \right)_{1a} (W_{sgoa} - W_{sl1a}) \quad (2-121)$$

$$\frac{d}{dt} P_{sgob} = \left( \frac{1}{ALC_2} \right)_{1b} (W_{sgob} - W_{sl1b}) \quad (2-122)$$

$$\frac{d}{dt} P_{sl1a} = \left( \frac{1}{ALC_2} \right)_{2a} (W_{sl1a} - W_{ra} - W_{sada} - W_{sl2a}) \quad (2-123)$$

$$\frac{d}{dt} P_{sl1b} = \left( \frac{1}{ALC_2} \right)_{2b} (W_{sl1b} - W_{rb} - W_{sadb} - W_{sl2b}) \quad (2-124)$$

$$\frac{d}{dt} P_{sl2a} = \left( \frac{1}{ALC_2} \right)_{3a} (W_{sl2a} - W_{mada} - W_{sl3a}) \quad (2-125)$$

$$\frac{d}{dt} P_{sl2b} = \left( \frac{1}{ALC_2} \right)_{3b} (W_{sl2a} - W_{madb} - W_{sl3b}) \quad (2-126)$$

$$\frac{d}{dt} P_t = \left( \frac{1}{ALC_2} \right)_t (W_{sl3a} + W_{sl3b} - W_t - W_{bp} - W_{rh2} - W_{pg}) \quad (2-127)$$

Neglecting the effect of kinetic energy change, the momentum equation is written for every junction between nodes as follows:

$$\frac{d}{dt} W_{sl1a} = \left( \frac{144g_c A}{L} \right)_{1a} (P_{sgoa} - P_{sl1a}) - K_{f1a} W_{sl1a} |W_{sl1a}| \quad (2-128)$$

$$\frac{d}{dt} W_{sl1b} = \left( \frac{144g_c A}{L} \right)_{1b} (P_{sgob} - P_{sl1b}) - K_{f1b} W_{sl1b} |W_{sl1b}| \quad (2-129)$$

$$\frac{d}{dt} W_{sl2a} = \left( \frac{144g_c A}{L} \right)_{2a} (P_{sl1a} - P_{sl2a}) - K_{f2a} W_{sl2a} |W_{sl2a}| \quad (2-130)$$

$$\frac{d}{dt} W_{sl2b} = \left( \frac{144g_c A}{L} \right)_{2b} (P_{sl1b} - P_{sl2b}) - K_{f2b} W_{sl2b} |W_{sl2b}| \quad (2-131)$$

$$\frac{d}{dt} W_{sl3a} = \left( \frac{144g_c}{L} \right)_{3a} (P_{sl2a} - P_t) - K_{f3a} W_{sl3a} |W_{sl3a}| \quad (2-132)$$

$$\frac{d}{dt} W_{sl3b} = \left( \frac{144g_c}{L} \right)_{3b} (P_{sl2b} - P_t) - K_{f3b} W_{sl3b} |W_{sl3b}| \quad (2-133)$$

The flows through the safety relief, atmospheric dump, turbine control, and bypass valves are assumed to be proportional to the pressure at the valve. All valve positions are given to the steam line model as boundary conditions.

$$W_{ra} = K_{ra} X_{ra} P_{sl1a} \quad (2-134)$$

$$W_{rb} = K_{rb} X_{rb} P_{sl1b} \quad (2-135)$$

$$W_t = K_t X_t P_t \quad (2-136)$$

$$W_{bp} = K_{bp} X_{bp} P_t \quad (2-137)$$

$$W_{sada} = K_{sada} X_{sada} P_{sl1a} \quad (2-138)$$

$$W_{sadb} = K_{sadb} X_{sadb} P_{sl1b} \quad (2-139)$$

$$W_{mada} = K_{mada} X_{mada} P_{sl2a} \quad (2-140)$$

$$W_{madb} = K_{madb} X_{madb} P_{sl2b} \quad (2-141)$$

#### Nomenclature

A	Flow area, ft <sup>2</sup>
1/ALC <sub>2</sub>	dP/dM for steam line nodes, psi/lbm
144g <sub>c</sub> A/L	Reciprocal of flow inertia for steam line nodes times gravitational conversion constant, lbm-in. <sup>2</sup> /lbf-s
C <sub>1</sub>	Density of steam at the reference pressure, lbm/ft <sup>3</sup>
C <sub>2</sub>	∂ρ/∂P for steam at the reference pressure, lbm-in. <sup>2</sup> /lbf-ft <sup>3</sup>
K	Proportionality constant for valve flows, lbm-in. <sup>2</sup> /lbf-s
K <sub>f</sub>	Friction loss proportionality constant for steam line nodes, 1/lbm
P	Pressure, psia
W	Mass flow rate, lbm/s

$W_{bp}$	Bypass valve flow rate, lbm/s
$W_{pg}$	Pegging steam flow rate, lbm/s
$W_{rh2}$	Reheater flow rate, lbm/s
$X$	Valve position, %/100%
$\rho$	Density, lbm/ft <sup>3</sup>

#### Subscripts

mada, madb	Main atmospheric dump valves (on/off)
ra, rb	Relief valves
sada, sadb	Small atmospheric dump valves (modulating)
sgia, sgib	Steam generator inlet
sgoa, sgob	Steam generator outlet
sl	Steam line node
1a,1b,2a,2b	Steam line node designations
3a,3b,t	

#### 2.19. Valves on Steam Lines

The response of the turbine throttle valve to the control signal supplied by the ICS is modeled according to equation 2-142:

$$\frac{dX_t}{dt} = (X_{TD} - X_t)/\tau. \quad (2-142)$$

The value of the derivative is limited to 10%/s opening velocity and 100%/s closing velocity. The throttle valve model is also used to model the turbine stop valves.

The main steam relief (safety) valves on each steam line are modeled as two banks, each bank with its own opening and closing pressure. For the purposes of simulation, all valves in a bank are combined; therefore, a total of four valves are simulated. Each valve steps fully open when the opening setpoint pressure is exceeded and steps fully closed when pressure drops below the closing setpoint pressure. The atmospheric dump valves comprise on-off and modulating valves. There is one modulating valve per steam line. The valve opens

and closes according to the position demand generated by the proportional-integral (PI) valve controller. The on-off atmospheric dump valves comprise four banks per steam line. These valves step fully open and fully closed after a delay time specified by the input data. The opening and closing setpoints of all four banks assume different values under different trip conditions. For example, the setpoints after a reactor trip are different from the normal ones.

The condenser dump valves are modeled as a single bank of modulating valves. The valve position opens and closes within rate limits according to the position demanded by a PI controller. In addition, the controller setpoint changes for different trip conditions in the same manner as the setpoints for the on-off atmospheric dumps.

#### Nomenclature

$X_t$	Turbine control valve position, %/100%
$X_{TD}$	Turbine control valve position demand, %/100%
$\tau$	Turbine control valve time constant, s

#### 2.20. Turbine and Associated Components

The turbine model includes the moisture separator and reheaters. The dynamics of the turbine model are approximated by applying time constants to steady-state heat balance data for the power plant. The heat balance data as functions of throttle flow, flow out of the second reheater, or flow into the low pressure turbine are supplied along with the time constants as part of the input data to the simulation. The extraction flows are shown in Figure 1-2. The equations that model the dynamics of these flows are given below. The B terms are logical variables having a value of either 1 or 0, depending on plant status. The F terms represent steady-state table data supplied as part of the input data. All flows are defined in Figure 1-2.

$$\frac{d}{dt} W_{6b} = \frac{1}{\tau_{6b}} [B_6 F_1(W_t) - W_{6b}] \quad (2-143)$$

$$\frac{d}{dt} W_{5b} = \frac{1}{\tau_{5b}} [B_5 F_2(W_t) - W_{5b}] \quad (2-144)$$

$$\frac{d}{dt} W_4 = \frac{1}{\tau_4} [B_4 F_3(W_t) - W_4] \quad (2-145)$$

$$\frac{d}{dt} W_{rh1} = \frac{1}{\tau_{grh}} [B_5 F_6 (W_t) - W_{rh1}] \quad (2-146)$$

$$\frac{d}{dt} W_7 = \frac{1}{\tau_7} [W_t - B_6 F_1 (W_t) - B_5 F_2 (W_t) - B_4 F_3 (W_t) - B_5 F_6 (W_t) - W_7] \quad (2-147)$$

$$W_{ms} = F_1 (W_7) \quad (2-148)$$

$$\frac{d}{dt} W_{sd} = \frac{1}{\tau_{rh1}} (W_7 - W_{ms} - W_{sd}) \quad (2-149)$$

$$\frac{d}{dt} W_{sc} = \frac{1}{\tau_{rh2}} (W_{sd} - W_{sc}) \quad (2-150)$$

$$W_{rh2} = B_6 F_1 (W_{sc}) \quad (2-151)$$

$$W_{sb} = F_2 (W_{sc}) \quad (2-152)$$

$$W_{sa} = W_{sc} - W_{sb} \quad (2-153)$$

$$W_s = B_{lpi} W_{sa} \quad (2-154)$$

$$\frac{d}{dt} W_3 = \frac{1}{\tau_3} [B_3 F_1 (W_s) - W_3] \quad (2-155)$$

$$\frac{d}{dt} W_2 = \frac{1}{\tau_2} [B_2 F_2 (W_s) - W_2] \quad (2-156)$$

$$\frac{d}{dt} W_1 = \frac{1}{\tau_1} [B_1 F_3 (W_s) - W_1] \quad (2-157)$$

$$\frac{d}{dt} W_9 = \frac{1}{\tau_9} [W_s - B_3 F_1 (W_s) - B_2 F_2 (W_s) - B_1 F_3 (W_s) - W_9] \quad (2-158)$$

#### Logical Variables

$$B_{lpi} = 0 \text{ when intercept valves are closed} \quad (2-159)$$

$$B_1 = 0 \text{ when heater 1 is failed} \quad (2-160)$$

$$B_2 = 0 \text{ when heater 2 is failed} \quad (2-161)$$

$$B_3 = 0 \text{ when heater 3 is failed} \quad (2-162)$$

$$B_4 = 0 \text{ when heater 4 is failed} \quad (2-163)$$

$$B_5 = 0 \text{ when heater 5 is failed} \quad (2-164)$$

$$B_6 = 0 \text{ when heater 6 is failed} \quad (2-165)$$

#### Tables of Flow at Steady State

$$F_1(W_t) = W_{6b} \text{ Vs } W_t \quad (2-166)$$

$$F_2(W_t) = W_{5b} \text{ Vs } W_t \quad (2-167)$$

$$F_3(W_t) = W_4 \text{ Vs } W_t \quad (2-168)$$

$$F_6(W_t) = W_{rh1} \text{ Vs } W_t \quad (2-169)$$

$$F_1(W_{8c}) = W_{rh2} \text{ Vs } W_{8c} \quad (2-170)$$

$$F_2(W_{8c}) = W_{8b} \text{ Vs } W_{8c} \quad (2-171)$$

$$F_1(W_8) = W_3 \text{ Vs } W_8 \quad (2-172)$$

$$F_2(W_8) = W_2 \text{ Vs } W_8 \quad (2-173)$$

$$F_3(W_8) = W_1 \text{ Vs } W_8 \quad (2-174)$$

#### 2.21. Generator

The simulation of the electric power generator is based on a torque balance approach. The steam throughput of the high- and low-pressure turbines is used to determine the net mechanical steam torque; the electrical load torque is determined from the generator load angle. Electrical damping, friction, and windage torques are also included. The generator frequency is calculated from the net torque. The grid frequency is allowed to vary from normal grid frequency in step fashion if the operator so desires. The equations describing the dynamic behavior of the generator are given below.

$$M_s = F_2(W_7) + F_1(W_9) \quad (2-175)$$

$$M_e = B_{tgc} [B_t K_{tg2} + K_{tg3} \int (F_g - F_{gr}) dt] \quad (2-176)$$

$$M_d = B_{tgc} [K_{tg4} (F_g - F_{gr})] \quad (2-177)$$

$$M_f = K_{tgs} F_g \quad (2-178)$$

$$M_g = M_s - M_e - M_d - M_f \quad (2-179)$$

$$\frac{d}{dt} F_g = K_{tg1} M_g \quad (2-180)$$

$$F_{gr} = F_{grn} + B_{fup} \Delta F_{gr} - B_{fdn} \Delta F_{gr} \quad (2-181)$$

### Nomenclature

$F_g$	Generator frequency, Hz
$F_{gr}$	Grid frequency, Hz
$\Delta F_{gr}$	Grid frequency step, Hz
$F_{grn}$	Normal grid frequency, Hz
$K_{tg1}$	Turbine-generator speed change rate at 100% torque, Hz/s
$K_{tg2}$	Normal house load as a fraction of total load, %/100%
$K_{tg3}$	Turbine-generator torque change per load angle change, %/deg
$K_{tg4}$	Turbine-generator damping constant, %/Hz
$K_{tg5}$	Turbine-generator friction and windage torque, %/Hz
$M_d$	Electrical damping torque, %/100%
$M_e$	Electrical load torque, %/100%
$M_f$	Friction and windage torque, %/100%
$M_g$	Net torque, %/100%
$M_s$	Mechanical steam torque, %/100%

### Logical Variables

$$B_{fdn} = 1 \text{ when pushbutton is depressed to step down grid frequency} \quad (2-182)$$

$$B_{fup} = 1 \text{ when pushbutton is depressed to step up grid frequency} \quad (2-183)$$

$$B_t = 0 \text{ when turbine is tripped} \quad (2-184)$$

$$B_{tgc} = 0 \text{ when grid breakers are open (loss of load)} \quad (2-185)$$

### Tables

$$F_2(W_7) = \text{High-pressure turbine steam torque Vs } W_7 \quad (2-186)$$

$$F_1(W_9) = \text{Low-pressure turbine steam torque Vs } W_9 \quad (2-187)$$

## 2.22. Feedwater Flows and Pressures

The feedwater and condensate system includes all the equipment that connects the condenser to the steam generator. It is much easier to discuss the total system if it is divided into separate parts. Therefore, in the following discussion of the model, the feedwater system is defined as the equipment on the downstream side of the main feedwater pumps. The condensate system is that equipment from the condenser to the main feedwater pumps. The emergency feedwater system and the total secondary energy balance are also discussed separately.

The feedwater system is modeled as a single train (see Figure 1-2). The calculation of the system flows and pressures is based on the assumption of constant feedwater densities in the various sections. Because of this assumption, the conservation of momentum is the only differential equation to be solved for flows and pressures. The calculation of feedwater enthalpies and temperatures is discussed in section 2.27.

The momentum equation for constant-area flow channels, after neglecting the pressure drop due to spatial acceleration (a result of the constant-density assumption), becomes

$$\frac{1}{A} \frac{\partial}{\partial t} W + 144g_c \frac{\partial}{\partial x} P + \frac{f/\rho}{2D_h A^2} |W|W + g\rho = 0. \quad (2-188)$$

It is further assumed that for a particular section of pipe the friction factor  $f$  is constant, and the differential pressure in the spatial dimension can be evaluated in the same manner as for the steam lines

$$\frac{\partial}{\partial x} P = \frac{P_o - P_i}{L} \quad (2-189)$$

where  $P_o$  is the pressure at the downstream node,  $P_i$  is the driving pressure, and  $L$  is the length of the section of pipe. Combining equations 2-188 and 2-189 and solving for the time rate of change of the flow rate results in the following general equation for the feedwater flow rates:

$$\frac{d}{dt} W = -\frac{144g_c A}{L} (P_i - P_o - \Delta P_f - \Delta P_e). \quad (2-190)$$

The elevation pressure drop  $\Delta P_e$  is a constant, and the frictional drop  $\Delta P_f$  is calculated from the following expression (note that flow is in the positive direction only; therefore  $|W|W = W^2$ ):

$$\Delta P_f = K_f W^2. \quad (2-191)$$

The friction coefficient  $K_f$  is assumed to be constant and is determined from the initial pressures and flows so that the flow rate derivative equals zero. In lines having control valves the frictional pressure drop across the valve is a function of valve position (section 2.23). In lines with feedwater pumps the driving pressure includes the pressure head developed by the pump (section 2.24). The following equations are used to calculate the feedwater flows to each steam generator and the flow through the main feedwater pumps:

$$\frac{d}{dt} W_{fwa} = \left( \frac{144 g_c A}{L} \right)_{fwa} (P_{fw} - P_{sgia} - \Delta P_{ea} - K_{fa} W_{fwa}^2 - \Delta P_{cva}), \quad (2-192)$$

$$\frac{d}{dt} W_{fwb} = \left( \frac{144 g_c A}{L} \right)_{fwb} (P_{fw} - P_{sgib} - \Delta P_{eb} - K_{fb} W_{fwb}^2 - \Delta P_{cvb}), \quad (2-193)$$

$$W_{fw} = W_{fwa} + W_{fwb}. \quad (2-194)$$

The pressure ( $P_{fw}$ ) at the common junction is found by solving the conservation of momentum equation for the main feedwater line and by differentiating equation 2-194:

$$\frac{d}{dt} W_{fw} = \frac{d}{dt} W_{fwa} + \frac{d}{dt} W_{fwb}, \quad (2-195)$$

$$P_{fw} = P_{c2} + \Delta P_{fp} + \Delta P_{efw} - K_{fw} W_{fw}^2 - \left( \frac{1}{144 g_c A} \right)_{fw} \frac{d}{dt} W_{fw}. \quad (2-196)$$

#### Nomenclature

A	Cross-sectional area, ft <sup>2</sup>
D <sub>h</sub>	Hydraulic diameter, ft
f	Friction factor, dimensionless
g	Gravitational acceleration, ft/s <sup>2</sup>
g <sub>c</sub>	Gravitational conversion constant, lbm-ft/lbf-s

$K_f$	Proportionality constant for friction pressure drop, $\text{psi}/(\text{lbm/s})^2$
$K_{fa}, K_{fb}$	Value of $K_f$ for feedwater piping from the common junction to steam generators A and B, $\text{psi}/(\text{lbm/s})^2$
$K_{fw}$	Value of $K_f$ for main feedwater line, $\text{psi}/(\text{lbm/s})^2$
$L$	Length of section of piping, ft
$(L/144g_c A)_{fw}$	Length over area term for the main feedwater piping, $(\text{lbm/s}^2)/\text{psi}$
$(144g_c A/L)_{fwa}$	Reciprocal of length over area term for the feedwater piping from the common junction to steam generator A, $\text{psi}/(\text{lbm/s}^2)$
$(144g_c A/L)_{fwb}$	Reciprocal of length over area term for the feedwater piping from the common junction to steam generator B, $\text{psi}/(\text{lbm/s}^2)$
$P$	Pressure, psia
$P_{c2}$	Pressure at feed pump inlet, psia
$P_{fw}$	Pressure at the common junction, psia
$P_1$	Inlet pressure, psia
$P_o$	Outlet pressure, psia
$P_{sgia}, P_{sgib}$	Pressures at secondary inlets to each steam generator, psia
$\Delta P_{cva}, \Delta P_{cvb}$	Pressure drop across control valves A and B, psi
$\Delta P_e$	Elevation pressure drop or gain, psi
$\Delta P_{ea}, \Delta P_{eb}$	Elevation pressure drop in feedwater lines from the common junction to steam generators A and B, psi
$\Delta P_{efw}$	Elevation pressure gain in main feedwater line from the feed pump inlet to the common junction, psi
$\Delta P_{fp}$	Feed pump pressure head, psi
$W$	Mass flow rate, lbm/s
$W_{fw}$	Mass flow rate into the common junction, lbm/s
$W_{fwa}, W_{fwb}$	Mass flow rates from the common junction to steam generators A and B, lbm/s

### 2.23. Feedwater Control Valves

Simulation of the feedwater control valves involves computation of the valve position, which is then used to calculate the valve coefficient ( $C_v$ ). The valve coefficient and feedwater flow through the valve are then used to calculate the pressure drop across the valve.

The  $C_v$  of a valve is defined as the flow of water at 60F (in gpm) that will result in a pressure drop of one psi across the valve:

$$\Delta P_v = \frac{\rho}{62.4} \left( \frac{Q}{C_v} \right)^2 \quad (2-197)$$

For flows given in lbm/s,

$$\Delta P_v = \frac{3228.34}{\rho} \left( \frac{W}{C_v} \right)^2 \quad (2-198)$$

The flow of feedwater to each steam generator during normal operation is controlled by two valves operating in parallel. The combined valve coefficient as a function of the percent opening of the valve is then used in equation 2-198 to find the valve pressure drop:

$$C_v = F(\%X_v). \quad (2-199)$$

The percent feedwater valve opening is calculated by a set of equations that simulate the feedwater valve motor speed and valve position. The control system supplies a feedwater flow error signal to the valve motor. If the feedwater flow needed is greater than that being supplied, the feedwater valve motor is started; this opens the valve. The valve continues to open until the flow error is brought to zero. The operation of the valve for situations where the feedwater flow required is less than that being supplied is analogous to the above except that the valve is closed. Whether the valve is opening or closing, it is limited to a maximum stroke rate regardless of the magnitude of the error. The operation of the feedwater control valves is described by the following equations (same form for both valves):

$$\frac{d}{dt} v_v = \frac{1}{\tau_v} (v_{vd} - v_v), \quad v_{\max} \geq v_v \geq -v_{\max} \quad (2-200)$$

$$\frac{d}{dt} X_v = v_v \quad (2-201)$$

## Nomenclature

$C_v$	Valve coefficient, $\text{gpm}/\sqrt{\text{psi}}$
$\Delta P_v$	Pressure drop across valve, psi
$Q$	Volumetric flow, gpm
$V_{\max}$	Maximum valve opening or closing speed, %/100%-s
$V_v$	Valve closing velocity, %/100%-s
$V_{vd}$	Valve closing velocity demand, %/100%-s
$W$	Mass flow rate, lbm/s
$\%X_v$	Valve position, %/100%
$\rho$	Density, $\text{lbm/ft}^3$
$\tau_v$	Valve time constant, s

## 2.24. Main Feedwater Pumps

The head developed by the main feedwater pumps is a function of the number of pumps running (normally two), the pump speed, the flow through each pump, and whether or not the pressure at the pump suction has dropped below the net positive suction head (NPSH) value.

The homologous pump curves are used to calculate the pump head (in psi). Pump speed ( $N_p$ ), flow ( $W_p$ ), and head ( $\Delta P_p$ ) are normalized to rated (normally 100%) values, and these normalized curves are then used to compute the head given flow and speed. Define

$$\alpha = N_p / N_r, \quad (2-202)$$

$$\beta = W_p / W_r, \quad (2-203)$$

$$\gamma = \Delta P_p / \Delta P_r. \quad (2-204)$$

Construct two tables at  $N_p = N_r$  ( $\alpha = 1.0$ ).

Table 1:  $W_p \leq W_r$

$$\beta/\alpha \text{ Vs } \gamma/\alpha^2 \quad (2-205)$$

Note:  $\beta/\alpha = W_p / W_r, \quad (2-206)$

$$\gamma/\alpha^2 = \Delta P_p / \Delta P_r, \quad (2-207)$$

Table 2:  $W_p \geq W_r$

$$\alpha/\beta \text{ Vs } \gamma/\beta^2 \quad (2-208)$$

Note:  $\alpha/\beta = W_r/W_p$ , (2-209)

$$\gamma/\beta^2 = (\Delta P_p / \Delta P_r) / (W_r / W_p)^2 \quad (2-210)$$

Now, given pump speed and pump flow for any operating condition, compute  $\alpha$  and  $\beta$ . If  $\alpha \geq \beta$ , use Table 1 to find  $(\gamma/\alpha^2)_1$ . Then  $\Delta P_p = (\gamma/\alpha^2)_1 \alpha^2 \Delta P_r$ . If  $\alpha < \beta$ , use Table 2 to find  $(\gamma/\beta^2)_2$ . Then  $\Delta P_p = (\gamma/\beta^2)_2 \beta^2 \Delta P_r$ .

The feedwater pump speed is calculated from a fit to manufacturer's data for the pump controller and is modeled as a second-order system. The pump speed ( $N_p$ ) in response to a speed demand signal ( $N_{pd}$ ) received from the control system is defined by equation 2-211:

$$\frac{d^2}{dt^2} N_p = \frac{1}{\tau_{p2}} N_{pd} - N_p - \tau_{p1} \frac{d}{dt} N_p. \quad (2-211)$$

If the suction pressure ( $P_{c2}$ ) drops below the minimum suction pressure (NPSH), the pump speed demand ( $N_{pd}$ ) in the equation above is set equal to zero.

#### Nomenclature

$N_p, N_r$	Actual and rated pump speed, rpm
$N_{pd}$	Pump speed demand, rpm
$\Delta P_p, \Delta P_r$	Actual and rated pump head, psi
$W_p, W_r$	Actual and rated pump flow, lbm/s
$\alpha$	Normalized pump speed (dimensionless)
$\beta$	Normalized pump flow (dimensionless)
$\gamma$	Normalized pump head (dimensionless)
$\tau_{p1}, \tau_{p2}$	Pump controller time constants, s

#### 2.25. Emergency Feedwater System

The emergency feedwater system provides for steam generator flow in the event of feedwater or steam line isolation. This would occur if all main feed pumps were tripped or if the steam line pressure fell below a setpoint pressure. When isolation occurs, block valves in the feedwater and steam lines are closed

(steam line isolation occurs only if steam pressure is low) and a delay timer is started to simulate startup of the emergency feedwater pumps. After the delay time has elapsed, emergency feedwater flow is provided to the steam generators. Emergency feedwater flow control is modeled as a proportional-integral controller that acts to maintain a constant steam generator level. In the POWER TRAIN simulation any isolation signal generated causes isolation of both steam generators.

## 2.26. Condensate System Flows and Pressures

The condensate system is composed of all the equipment from the condenser to the main feedwater pumps. It is shown in schematic form in Figure 1-2. This section deals with the calculation of the various flow rates and pressures in the system and with the calculation of the hot well level. Refer to Figure 1-2 for definition of all flows and pressures. The flows and pressures are calculated using the same assumptions and approach as outlined in section 2.22. The main system flows are calculated from the main feedwater flow and the drain tank flows that are pumped forward.

$$W_{c2} = W_{fw} - W_{dhp} \quad (2-212)$$

$$W_{c1} = W_{c2} - W_{d\&p} \quad (2-213)$$

The drain tank flows are calculated assuming that a constant drain tank level exists and that the flow rate differential may be approximated as a simple first-order lag. Capability is also provided to terminate drain tank flow instantly, which simulates closure of a block valve. If block valves are open,

$$\frac{d}{dt} W_{dhp} = \frac{1}{\tau_{dhp}} (W_{4d} + W_{5d} + W_{ms} - W_{dhp}) \quad (2-214)$$

and

$$\frac{d}{dt} W_{d\&p} = \frac{1}{\tau_{d\&p}} (W_{1d} + W_{fpt} - W_{d\&p}). \quad (2-215)$$

If the high-pressure drain tank block valve closes,

$$\frac{d}{dt} W_{dhp} = 0, \quad W_{dhp} = 0. \quad (2-216)$$

If the low-pressure drain tank block valve closes,

$$\frac{d}{dt} W_{d\&p} = 0, \quad W_{d\&p} = 0. \quad (2-217)$$

Now all the system flow rates and their derivatives are defined, so it is possible to solve the momentum equation for each line to determine the pressures. From equations 2-212 and 2-213, it follows that

$$\frac{d}{dt} W_{c2} = \frac{d}{dt} W_{fw} - \frac{d}{dt} W_{dhp} \quad (2-218)$$

and

$$\frac{d}{dt} W_{c1} = \frac{d}{dt} W_{c2} - \frac{d}{dt} W_{d\&p}. \quad (2-219)$$

Therefore,

$$P_{c1} = P_c + \Delta P_{cp} + \Delta P_{hwp} - \Delta P_{ec1} + \Delta P_{hw\&l} - K_{fc1} W_{c1}^2 - \left( \frac{L}{144 g_c A} \right)_{c1} \frac{d}{dt} W_{c1}, \quad (2-220)$$

and

$$P_{c2} = P_{c1} - \Delta P_{ec2} - K_{fc2} W_{c2}^2 - \left( \frac{L}{144 g_c A} \right)_{c2} \frac{d}{dt} W_{c2}. \quad (2-221)$$

The head developed by the condensate and hot well pumps depends on the number of pumps running, the flow rate through the pumps, and the pump head capacity curves:

$$\Delta P_{cp} = F_{cp} (W_{c1} / N_{cp}), \quad (2-222)$$

$$\Delta P_{hwp} = F_{hwp} (W_{c1} / N_{vp}). \quad (2-223)$$

The hot well level is determined from a mass balance on the condenser:

$$\frac{d}{dt} L_{hw} = \left( \frac{1}{A_p} \right)_{hw} (W_g + W_{bp} - W_{c1}). \quad (2-224)$$

The pressure due to the hot well level is found from the initial pressure and level, which are input, and the level calculated from equation 2-224:

$$\Delta P_{hw\ell} = \left( \frac{P_{hw\ell}}{L_{hw}} \right)_{t=0} L_{hw} \quad (2-225)$$

Calculation of the drain flows for each heater is discussed in the next section, which covers the energy balance for the entire feedwater and condensate system.

#### Nomenclature

$(1/A\rho)_{hw}$	Reciprocal of hot well area times density, ft/lbm
$F_{cp}$	Condensate pump head versus flow table, psi
$F_{hwp}$	Hot well pump head versus flow table, psi
$K_{fc1}$	Proportionality constant for friction pressure drop between condenser and $P_{c1}$ , psi/(lbm/s) <sup>2</sup>
$K_{fc2}$	Proportionality constant for friction pressure drop between $P_{c1}$ and $P_{c2}$ , psi/(lbm/s) <sup>2</sup>
$L_{hw}$	Hot well level, ft
$(L/144g_c A)_{c1}$	Length over area term for piping between condenser and $P_{c1}$ , lbm/s <sup>2</sup> -psi
$(L/144g_c A)_{c2}$	Length over area term for piping between $P_{c1}$ and $P_{c2}$ , lbm/s <sup>2</sup> -psi
$\Delta P_{cp}$	Condensate pump head, psi
$\Delta P_{ec1}$	Elevation pressure difference between condenser and $P_{c1}$ , psi
$\Delta P_{ec2}$	Elevation pressure difference between $P_{c1}$ and $P_{c2}$ , psi
$\Delta P_{hw}$	Pressure difference caused by hot well level, psi
$\Delta P_{hwp}$	Hot well pump head, psi
$(\Delta P/L_{hw})_{t=0}$	Initial value of hot well pressure difference divided by initial hot well level, psi/ft
$\tau_{dhp}$	High-pressure drain tank time constant, s
$\tau_{d\&p}$	Low-pressure drain tank time constant, s

#### 2.27. Secondary System Energy Balance

The secondary system energy balance is based on plant heat balance information. Several functions of load are defined as part of the input data. Differential and algebraic equations are solved after several system simplifications are made.

As an example, the actual plant has parallel feedwater heaters. In this model they are lumped into one heater. Further, it is assumed that the high-pressure heaters (if they are not failed) will provide feedwater at the proper temperature, depending on load. To accomplish this, the necessary steam flow to the high-pressure heaters is calculated based on the system energy balance. This simulates the increased shell-side flow that would occur due to a lower shell-side pressure (which would result from introduction of feedwater at a lower temperature into the heater).

The basic method is to write steady-state mass and energy balances for each heater and then to assume that the transient performance of each heater can be approximated as a first-order system. The characteristic time constant of the heater is used for both the mass and energy balances. With reference to Figure 1-2, the following equations are presented (beginning at the steam generators and working back to the condenser).

#### 2.27.1. Feedwater Temperatures and Delays

$$T_{fwa}(t) = T_{fw}[t - (M_{fwa}/W_{fwa})] \quad (2-226)$$

$$T_{fwb}(t) = T_{fw}[t - (M_{fwb}/W_{fwb})] \quad (2-227)$$

$$T'_{fw}(t) = T'_{fw}[t - (M_{fw}/W_{fw})] \quad (2-228)$$

$$T'_{fw} = f(h_{hp6}) \quad (2-229)$$

#### 2.27.2. High-Pressure Heater 6

$$h_{hp6i} = h_{hp6i}(t) = h_{hp5}[t - (M_{fws}/W_{fw})] \quad (2-230)$$

$$h_{6d} = F_1(W_{fw}) \quad (2-231)$$

$$W_{pg6} = [W_{fw}(h_{hp6} - h_{hp6i}) + W_{sd}h_{6d} - W_{sb}h_{6b} - W_{rh2}h_{rh2d}]/h_{pg6} \quad (2-232)$$

If  $W_{pg6} > W_{pg6m}$ ,

$$W_{pg6} = W_{pg6m} \quad (2-233)$$

$$W_6 = W_{pg6} + W_{sb} - W_{rh2}, \quad (2-234)$$

and

$$h'_{hp6} = h_{hp6i} + (W_6 h_6 - W_{6d} h_{6d})/W_{fw}. \quad (2-235)$$

If  $W_{pg6} \leq W_{pgsm}$ ,

$$W_6 = [W_{fw}(h_{hp6} - h_{hp6i}) + W_{6d}h_{6d}]/h_6 \quad (2-236)$$

and

$$h'_{hp6} = B_6 F_4(W_t) + \bar{B}_6 h_{hp6i} \quad (2-237)$$

If  $W_{pg6} < 0$ ,

$$W_{pg6} = 0 \quad (2-238)$$

and

$$W_6 + W_{6b} + W_{rh2} \quad (2-239)$$

Finally, regardless of the value of  $W_{pg6}$ ,

$$\frac{d}{dt} h_{hp6} = \frac{1}{\tau_{hp6}} (h'_{hp6} - h_{hp6}) \quad (2-240)$$

and

$$\frac{d}{dt} W_{6d} = \frac{1}{\tau_{hp6}} (W_6 - W_{6d}) \quad (2-241)$$

### 2.27.3. High-Pressure Heater 5

$$h_{hps1} = h_{hps1}(t) = h_{hc2}[t - (M_{fw4}/W_{fw})] \quad (2-242)$$

$$h_{5d} = F_2(W_{fw}) \quad (2-243)$$

$$h_{rh1d} = F_1(W_{8d}) \quad (2-244)$$

$$W_{pg5} = [W_{fw}(h_{hps} - h_{hps1} - \Delta h_{fp}) + W_{5d}h_{5d} - W_{6d}h_{6d} - W_{5b}h_{5b} - W_{rh1}h_{rh1d}]/h_{pg5} \quad (2-245)$$

If  $W_{pg5} > W_{pgsm}$ ,

$$W_{pg5} = W_{pgsm} \quad (2-246)$$

$$W_5 = W_{pg5} + W_{5b} + W_{rh1} \quad (2-247)$$

and

$$h'_{hps} = h_{hps1} + \Delta h_{fp} + (W_5 h_5 + W_{6d}h_{6d} - W_{5d}h_{5d})/W_{fw} \quad (2-248)$$

If  $W_{pg5} \leq W_{pg5m}$ ,

$$W_s = [W_{fw}(h_{hp5} - h_{hp5i} - \Delta h_{fp}) + W_{sd}h_{sd} - W_{6d}h_{6d}]/h_s \quad (2-249)$$

and

$$h'_{hp5} = B_5 F_5 (W_t) + \bar{B}_5 [h_{hp5i} + \Delta h_{fp}]. \quad (2-250)$$

If  $W_{pg5} < 0$ ,

$$W_{pg5} = 0 \quad (2-251)$$

and

$$W_s = W_{sb} + W_{rh1}. \quad (2-252)$$

Finally, regardless of the value of  $W_{pg5}$ ,

$$\frac{d}{dt} h_{hp5} = \frac{1}{\tau_{hp5}} (h'_{hp5} - h_{hp5}) \quad (2-253)$$

and

$$\frac{d}{dt} W_{sd} = \frac{1}{\tau_{hp5}} (W_s + W_{sd} - W_{sd}). \quad (2-254)$$

#### 2.27.4. High-Pressure Heater 4

$$h_{4d} = F_1 (W_{c2}) \quad (2-255)$$

$$h_{hp4i} = h_{hp4i}(t) = h_{4p3} [t - (M_{fw3}/W_{c2})] \quad (2-256)$$

$$h'_{hp4} = B_4 [(W_{c2} h_{hp4i} + W_4 h_4 - W_{4d} h_{4d})/W_{c2}] + \bar{B}_4 h_{hp4i} \quad (2-257)$$

$$\frac{d}{dt} h_{hp4} = \frac{1}{\tau_{hp4}} (h'_{hp4} - h_{hp4}) \quad (2-258)$$

$$\frac{d}{dt} W_{4d} = \frac{1}{\tau_{hp4}} (W_4 - W_{4d}) \quad (2-259)$$

#### 2.27.5. High-Pressure Heater Drain Tank

$$h_{ms} = F_3 (W_7) \quad (2-260)$$

$$h'_{dhp} = \frac{W_{sd}h_{sd} + W_{4d}h_{4d} + W_{ms}h_{ms}}{W_{sd} + W_{4d} + W_{ms}} \quad (2-261)$$

$$\frac{d}{dt} h_{dhp} = \frac{1}{\tau_{dhp}} (h'_{dhp} - h_{dhp}) \quad (2-262)$$

$$h_{hc2} = (W_{dhp} h_{dhp} + W_{c2} h_{hp4}) / W_{fw} \quad (2-263)$$

#### 2.27.6. Low-Pressure Heater 3

$$h_{lp3i} = h_{lp3i}(t) = h_{lp2} [t - (M_{fw2} / W_{c3})] \quad (2-264)$$

$$h'_{lp3} = B_3 [W_{c2} h_{lp3i} + W_3 h_3 - W_{3d} h_{3d}] / W_{c2} + \bar{B}_3 h_{lp3i} \quad (2-265)$$

$$h_{3d} = F_2 (W_{c2}) \quad (2-266)$$

$$\frac{d}{dt} h_{lp3} = \frac{1}{\tau_{lp3}} (h'_{lp3} - h_{lp3}) \quad (2-267)$$

$$\frac{d}{dt} W_{3d} = \frac{1}{\tau_{lp3}} (W_3 - W_{3d}) \quad (2-268)$$

#### 2.27.7. Low-Pressure Heater 2

$$h_{lp2i} = h_{lp2i}(t) = h_{c1} [t - (M_{fw1} / W_{c2})] \quad (2-269)$$

$$h'_{lp2} = B_2 [(W_{c2} h_{lp2i} + W_2 h_2 + W_{3d} h_{3d} - W_{2d} h_{2d}) / W_{c2}] + \bar{B}_2 h_{lp2i} \quad (2-270)$$

$$h_{2d} = F_3 (W_{c2}) \quad (2-271)$$

$$\frac{d}{dt} h_{lp2} = \frac{1}{\tau_{lp2}} (h'_{lp2} - h_{lp2}) \quad (2-272)$$

$$\frac{d}{dt} W_{2d} = \frac{1}{\tau_{lp2}} (W_2 + W_{3d} - W_{2d}) \quad (2-273)$$

#### 2.27.8. Low-Pressure Heater 1

$$h_{1D} = F_1 (W_{c1}) \quad (2-274)$$

$$h_c = F_2 (W_{c1}) \quad (2-275)$$

$$h'_{lp1} = B_1 [(W_{c1} h_{c1} + W_1 h_1 + W_{2d} h_{2d} - W_{1d} h_{1d}) / W_{c1}] + \bar{B}_1 h_c \quad (2-276)$$

$$\frac{d}{dt} h_{lp1} = \frac{1}{\tau_{lp1}} (h'_{lp1} - h_{lp1}) \quad (2-277)$$

$$\frac{d}{dt} W_{1d} = \frac{1}{\tau_{lp1}} (W_1 + W_{2d} - W_{1d}) \quad (2-278)$$

#### 2.27.9. Low-Pressure Heater Drain Tank

$$h_{fptc} = F_7(W_t) \quad (2-279)$$

$$\frac{d}{dt} W_{fpt} = \frac{1}{\tau_{fpt}} (W_{sb} - W_{fpt}) \quad (2-280)$$

$$h'_{d\&p} = \frac{W_{1d}h_{1d} + W_{fpt}h_{fptc}}{W_{1d} + W_{fpt}} \quad (2-281)$$

$$\frac{d}{dt} h_{d\&p} = \frac{1}{\tau_{d\&p}} (h'_{d\&p} - h_{d\&p}) \quad (2-282)$$

$$h_{c1} = (W_{d\&p}h_{d\&p} + W_{c1}h_{lp1})/W_{c2} \quad (2-283)$$

#### Nomenclature

Symbols beginning with the letters P, W, h, and T correspond to pressures, mass flow rates, enthalpies, and temperatures, respectively, as shown in Figure 1-2.

$\Delta H_{fp}$	Enthalpy rise across main feedwater pumps, Btu/lbm
$M_{fw1}$	Liquid mass between feedwater heaters 1 and 2, lbm
$M_{fw2}$	Liquid mass between feedwater heaters 2 and 3, lbm
$M_{fw3}$	Liquid mass between feedwater heaters 3 and 4, lbm
$M_{fw4}$	Liquid mass between feedwater heaters 4 and 5, lbm
$M_{fw5}$	Liquid mass between feedwater heaters 5 and 6, lbm
$M_{fw}$	Liquid mass between feedwater heater 6 and the common junction, lbm
$M_{fwa}$	Liquid mass between the common junction and OTSG-A, lbm
$M_{fwb}$	Liquid mass between the common junction and OTSG-B, lbm
$W_{pgsm}, W_{pgsm}$	Maximum pegging steam flow rate, lbm/s
$\tau_{lp1}, \tau_{lp2}, \tau_{lp3}$	Low-pressure feedwater heater time constants, s
$\tau_{hp4}, \tau_{hp5}$	High-pressure feedwater heater time constants, s

$\tau_{dhp}$  High-pressure drain tank time constant, s

$\tau_{dlp}$  Low-pressure drain tank time constant, s

Logical Variables

$B_1 = 0$  when heater 1 is failed (2-284)

$B_2 = 0$  when heater 2 is failed (2-285)

$B_3 = 0$  when heater 3 is failed (2-286)

$B_4 = 0$  when heater 4 is failed (2-287)

$B_5 = 0$  when heater 5 is failed (2-288)

Tables of Steady-State Data

$F_1(W_{c1}) = h_{1d} \text{ Vs } W_{c1}$  (2-289)

$F_1(W_{c2}) = h_{4d} \text{ Vs } W_{c2}$  (2-290)

$F_2(W_{c2}) = h_{3d} \text{ Vs } W_{c2}$  (2-291)

$F_3(W_{c2}) = h_{2d} \text{ Vs } W_{c2}$  (2-292)

$F_1(W_{fw}) = h_{6d} \text{ Vs } W_{fw}$  (2-293)

$F_2(W_{fw}) = h_{5d} \text{ Vs } W_{fw}$  (2-294)

$F_3(W_t) = W_4 \text{ Vs } W_t$  (2-295)

$F_4(W_t) = h_{hp6} \text{ Vs } W_t$  (2-296)

$F_5(W_t) = h_{hps} \text{ Vs } W_t$  (2-297)

$F_7(W_t) = h_{fpt} \text{ Vs } W_t$  (2-298)

$F_3(W_u) = h_{ms} \text{ Vs } W_7$  (2-299)

$F_1(W_8) = W_3 \text{ Vs } W_8$  (2-300)

$F_2(W_8) = W_2 \text{ Vs } W_8$  (2-301)

$F_3(W_8) = W_1 \text{ Vs } W_8$  (2-302)

$F_1(W_{8d}) = h_{rhd} \text{ Vs } W_{8d}$  (2-303)

## 2.28. Secondary System Option

Nearly every nuclear power station has some differences in its secondary system. The system described in sections 2.26 and 2.27 can be considered typical and is based on the TVA Bellefonte design. A secondary system option is available which slightly modifies the system to represent the SMUD Rancho Seco secondary system.

The SMUD unit differs from TVA in that it does not have a low-pressure drain tank; drain flows from the feedwater pump turbine and low-pressure heater 1 are simply bypassed directly to the condenser. In addition to this, the drain flow from high-pressure heater 5 and the moisture separator flows are directed to high-pressure heater 4 rather than directly to the high-pressure drain tank. These minor differences result in the following changes to the equations presented in sections 2.26 and 2.27:

Equation 2-213

$$W_{c1} = W_{c2} \quad (2-304)$$

Equation 2-214

$$\frac{d}{dt} W_{dhp} = \frac{1}{\tau_{dhp}} (W_{4d} - W_{dhp}) \quad (2-305)$$

Equation 2-215 is eliminated and 2-217 applies all the time.

Equation 2-224

$$\frac{d}{dt} L_{hw} = \left( \frac{1}{A_p} \right)_{hw} (W_g + W_{bp} + W_{1d} + W_{fpt} - W_{c1}) \quad (2-306)$$

Equation 2-257

$$h'_{hp4} = B_4 [(W_{c2} h_{hp4i} + W_4 h_4 + W_{ms} h_{ms} + W_{5d} h_{5d} - W_{4d} h_{4d}) / W_{c2}] + \bar{B}_4 h_{hp4i} \quad (2-307)$$

Equation 2-259

$$\frac{d}{dt} W_{4d} = \frac{1}{\tau_{hp4}} (W_4 + W_{ms} + W_{5d} - W_{4d}) \quad (2-308)$$

Equation 2-261

$$h'_{hdp} = h_{4d} \quad (2-309)$$

Equations 2-281 and 2-282 eliminated.

Equation 2-283

$$h_{ci} = h_{lp1} \quad (2-310)$$

### 2.29. Integrated Control System

The POWER TRAIN model is controlled by a digital simulation of the integrated control system (ICS). The ICS is modeled using a simulation language that provides a number of operational elements, such as those found in a typical ICS. The ICS model is formed by specifying the interconnection of these various elements to represent the controllers and trip modules of a typical ICS. The ICS model is extensive, including all the major subsystems

- Unit load demand subsystem.
- Integrated master control subsystem.
- Steam generator feedwater control subsystem.
- Reactor control subsystem.

The ICS model produces demand signals for the reactor and control rod drive (CRD) models, the steam generator (feedwater) model, and the turbine pressure model. The demand signals are driven by a unit load demand (ULD) signal established by the operator. The ULD station allows ramps and steps of varying degrees, automatic demand limits, and runbacks under conditions when generation is limited. The digital simulation of the ICS handles both automatic coordination of the various demand signals and manual control of the signals by the operator. The ICS simulation has the capability to automatically establish a tracking mode, whenever conditions arise that would cause unit tracking, and to perform automatic demand runback and transfer of control signals when the tracking mode is established. The model operator has full control over the ICS simulation from the analog console and can sequence and initiate the desired combination of transients by pushing the appropriate buttons and entering parameters on the keyboard. The model is designed to perform all transients without restoring to modifications in the hybrid program.

Trips and abnormal conditions that can be initiated from the mini panel include the following:

- Manual reactor trip.
- RC pump trips (all pumps, one pump, or two pumps).
- Feedwater pump trip (one or two).
- Turbine trip.
- Loss of electrical load.
- Pressurizer heater failure.
- Enable/disable flux/flow trip of reactor.
- Enable/disable high primary pressure trip of reactor.
- Manual failure of drain and condensate pumps in feedwater train.

In addition, the operator of the simulation may select, via the mini panel, either manual or automatic control of the following controllers and final control elements:

- Turbine bypass valves.
- Turbine control valves.
- Reactor-steam generator demand.
- Feed valve demand (A or B).
- Feed valve position (A or B).
- Reactor demand.
- Control rod position.
- Unit load demand.

#### 2.30. Summary of Limitations of POWER TRAIN Simulation

The POWER TRAIN program was developed specifically for analysis of the dynamic response of the B&W nuclear power station for both normal and off-normal operating conditions. Typical types of analysis performed with POWER TRAIN (not necessarily limited to) are given below.

1. Control system optimization and evaluation of new control concepts.
2. Control system failure and system effects analysis.
3. Protection system (trip) failures and system effects analysis.
4. Verification of warranted plant maneuvering capability.
5. System analysis and scoping studies for many anticipated transients, e.g.,

- a. Turbine trip with and without reactor trip.
- b. Load rejection.
- c. Grid frequency upsets.
- d. Loss of feedwater (complete or partial).
- e. Loss of feedwater heaters.
- f. Condensate system failures.
- g. Spurious reactor trip.
- h. Primary pump trips.

The scaling requirements of the analog portion of the simulation result in the following limits on the magnitude and rate-of-change of the variables specified below (if these limits are exceeded, the code results are not valid):

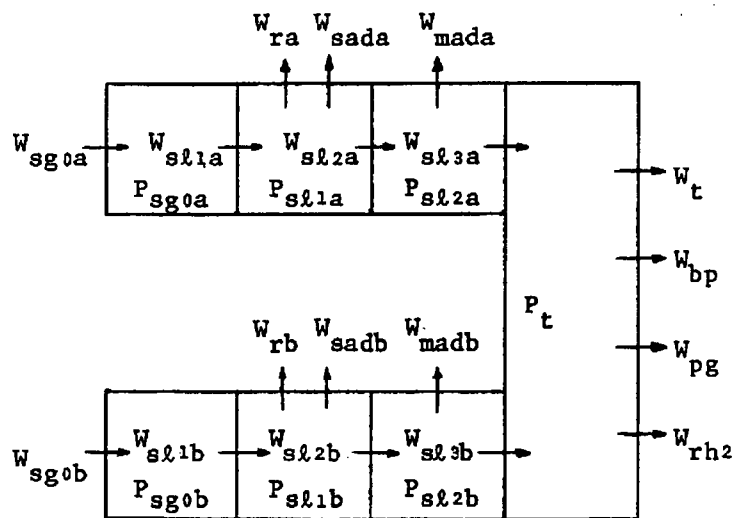
- Fluid temperatures: 350 to 650F
- Cladding and fuel temperatures: 350 to 3000 psia
- Primary flow rate: 5 to 200%
- Primary system pressure: 1500 to 3000 psia
- Reactor power level: 3 to 120%
- Secondary fluid enthalpy: 300 to 1400 Btu/lb
- Secondary fluid density: 0 to 60 lb/ft<sup>3</sup>
- Secondary flow rate: 0 to 115%
- Secondary pressure: 500 to 1500 psia
- Secondary pressure maximum rate-of-change: 100 psia/s
- Secondary flow rate maximum rate-of-change: 100%/s

In addition to the limits above, limitations on the maximum rate of change of the enthalpy increase on the secondary side of the steam generator simulation will result in a higher than normal heat removal rate for zero secondary flow conditions. Therefore, when the steam generator is dry, caution should be exercised in the interpretation of code results. It should always be kept in mind that this version of POWER TRAIN was developed as a power range (15-100%) analysis tool and is not suitable for analysis of low power (<15%) operation (e.g., emergency feedwater control studies) despite the fact that it will operate at low power levels.

Several other code limitations apply and are listed below. These limitations are necessary because of mathematical model assumptions, scaling limitations on the analog portion of the simulation, and the time step employed.

1. The pressurizer cannot go solid or completely empty. If either condition occurs, the code results from that point are invalid.
2. Two-phase conditions in the primary system are not modeled.
3. Operation of the secondary system which results in the introduction of saturated fluid conditions in the steam lines is not valid.
4. The code is not capable of analyzing system piping breaks in the primary system.
5. No emergency safety system features (such as high-pressure injection) are included in the code capability.
6. It is not valid for analysis of fast primary system reactivity excursions (e.g., rod ejections).

Figure 2-1. Steam Line Nodalization



### 3. INPUT/OUTPUT

#### 3.1. Input Data

Input data are required by each computer involved in the POWER TRAIN simulation. Input for POWER TRAIN consists of the following:

1. BLOCK DATA variables — Data for the portion of the simulation implemented on the CDC 1700 are input via a BLOCK DATA subprogram. All BLOCK DATA variables are defined in Table 3-1.
2. Tabular data — Tabular data required by the simulation are loaded onto the array processor via a program executed on the 1700. Tabular data are defined in Table 3-2.
3. Potentiometer setting data — COMANCHE, a program executed on the CDC 1700, sets all potentiometers based on user-supplied parameters. The parameters needed by COMANCHE are defined in Table 3-3.
4. ICS simulation — The ICS is simulated on the EAI-640 using the ISL simulation language. The use of ISL is defined in reference 1.

#### 3.2. Output Data

Non-real-time output, such as a dump of input data, is printed on the teletype; real-time output is plotted on strip chart recorders. Table 3-4 lists the simulation variables the user may specify to be plotted. In addition to those variables, any variable calculated on either analog computer may be plotted by directly patching the variable to the strip chart recorder. Some commonly used analog variables are listed in Table 3-5.

Table 3-1. BLOCK DATA Variables

<u>No.</u>	<u>Name</u>	<u>Description</u>
1	ALPHD	Doppler coefficient, $\Delta k/k/^{\circ}F$
2	ALPHM	Moderator coefficient, $\Delta k/k/^{\circ}F$
3	BETA(1)	Delayed neutron fraction, group 1
4	BETA(2)	Delayed neutron fraction, group 2
5	BETA(3)	Delayed neutron fraction, group 3
6	BETA(4)	Delayed neutron fraction, group 4
7	BETA(5)	Delayed neutron fraction, group 5
8	BETA(6)	Delayed neutron fraction, group 6
9	BETA(7)	Total delayed neutron fraction
10	CLAM(1)	Decay constant, group 1, $s^{-1}$
11	CLAM(2)	Decay constant, group 2, $s^{-1}$
12	CLAM(3)	Decay constant, group 3, $s^{-1}$
13	CLAM(4)	Decay constant, group 4, $s^{-1}$
14	CLAM(5)	Decay constant, group 5, $s^{-1}$
15	CLAM(6)	Decay constant, group 6, $s^{-1}$
16	GPCM	Cladding mass $\times$ cladding specific heat, Btu/ $^{\circ}F$
17	CPFM	Fuel mass $\times$ water specific heat, Btu/ $^{\circ}F$
18	CPWM	Water mass $\times$ water specific heat, Btu/ $^{\circ}F$
19		Not used
20		Not used
21	POWERI	Initial power, %
22	RHOALP	Fluid density, SG A lower plenum, lb/ft <sup>3</sup>
23	RHOAUP	Fluid density, SG A upper plenum, lb/ft <sup>3</sup>
24	RHOBLP	Fluid density, SG B lower plenum, lb/ft <sup>3</sup>
25	RHO BUP	Fluid density, SG B upper plenum, lb/ft <sup>3</sup>
26	RHOCLA	Fluid density, cold leg A, lb/ft <sup>3</sup>
27	RHOCLB	Fluid density, cold leg B, lb/ft <sup>3</sup>
28	RHOHLA	Fluid density, hot leg A, lb/ft <sup>3</sup>
29	RHOHLB	Fluid density, hot leg B, lb/ft <sup>3</sup>
30	RHOLP	Fluid density, vessel upper plenum, lb/ft <sup>3</sup>
31	RHOUP	Fluid density, vessel lower plenum, lb/ft <sup>3</sup>
32	RRM	Maximum rod worth, $\Delta k/k$
33	RSRM	Maximum safety rod worth, $\Delta k/k$

Table 3-2. (Cont'd)

<u>No.</u>	<u>Name</u>	<u>Description</u>
34	TAUDK	Decay heat time constant, s
35	TAUM	Rod drive time constant, s
36	TAUTS	Avg temp thermal time constant, s
37	TCIN	Initial cladding temperature F,
38	TFIN	Initial fuel temperature, F
39	TWIN	Initial coolant temperature, F
40	TWIIN	Initial core inlet temperature, F
41	VCORE	Core coolant volume, ft <sup>3</sup>
42	VCLA	Cold leg A volume, ft <sup>3</sup>
43	VCLB	Cold leg B volume, ft <sup>3</sup>
44	VHLA	Hot leg A volume, ft <sup>3</sup>
45	VHLB	Hot leg B volume, ft <sup>3</sup>
46	VLP	Vessel lower plenum volume, ft <sup>3</sup>
47	VUP	Vessel upper plenum volume, ft <sup>3</sup>
48	VSGA	SG A primary side volume, ft <sup>3</sup>
49	VSGB	SG B primary side volume, ft <sup>3</sup>
50	VSGALP	SG A lower plenum volume, ft <sup>3</sup>
51	VSGAUP	SG A upper plenum volume, ft <sup>3</sup>
52	VSGBLP	SG B lower plenum volume, ft <sup>3</sup>
53	VSGBUP	SG B upper plenum volume, ft <sup>3</sup>
54		Not used
55	WCBP	Core bypass flow rate, pps
56	WPA	Loop A flow rate, pps
57	WPB	Loop B flow rate, pps
58	XMAX	Maximum rod insertion, in.
59	XRO	Initial rod position, % withdrawn
60	VRS	Normal rod velocity, in./s
61	SGSIZE	No. of sample points in steam generators
62	K201	Rate of change of primary fluid density with pressure at constant temperature, lb/ft <sup>3</sup> -psia; use is optional
63	K202	Not used
64	K203	Level to volume equivalent in cylindrical section of pressurizer, in./ft <sup>3</sup>

Table 3-1. (Cont'd)

<u>No.</u>	<u>Name</u>	<u>Description</u>
65	K204	Level equivalent of volume of pressurizer below lower sensing tap, ft <sup>3</sup>
66	K205	Total pressurizer volume, ft <sup>3</sup>
67	K206	Equivalent coefficient for boiloff in pressurizer, lb/s-°F
68	K207	Not used
69	K208	Pressurizer wall temperature, F
70	K209	Low level cut-off for pressurizer heaters, in.
71	K210	Cut-on point for pressurizer heater 1, psia
72	K211	Proportional gain for variable output of heater 1, Btu/s-psia
73	K212	Low limit of prop. band of heater 1, psia
74	K213	Maximum capacity of heater 1, Btu/s
75	K214	Heater 2 capacity, Btu/s
76	K215	Heater 2 cut-on point, psia
77	K216	Heater 2 cut-off point, psia
78	K217	Heater 3 capacity, Btu/s
79	K218	Heater 3 cut-on point, psia
80	K219	Heater 3 cut-off point, psia
81	K220	Heater 4 capacity, Btu/s
82	K221	Heater 4 cut-on point, psia
83	K222	Pressurizer code safety valve flow rate, pps
84	K223	Static head, RC surge tap to pressurizer, psia
85	K224	Pressurizer code safety cut-on point, psia
86	K225	Pressurizer code safety cut-off point, psia
87	K226	Equivalent dynamic $\Delta P$ in surge line, psi/(pps) <sup>2</sup>
88	K227	Conversion factor for converting level error to makeup flow demand, pps/in.
89	K228	Proportional gain for level error, dimensionless
90	K229	Pressurizer level setpoint, in.
91	K230	Integral gain for level error, s <sup>-1</sup>
92	K231	Maximum RC makeup flow, pps
93	K232	Heater 4 cut-off point, psia
94	K233	Maximum spray flow, pps
95	K234	Spray cut-on point, psia

Table 3-1. (Cont'd)

<u>No.</u>	<u>Name</u>	<u>Description</u>
96	K235	Spray cut-off point, psia
97	K236	Letdown flow rate, pps
98	K237	Heater 1 cutoff point, psia
99	K238	Power-operated relief valve flow, pps
100	K239	Power-operated relief cut-on point, psia
101	K240	Power-operated relief cut-off point, psia
102	K241	Pressurizer heater time constant, s
103	K242	Pressurizer spray valve velocity, % stroke/s
104	PLAMBA	Mixing factor for pressurizer lower liquid region, s <sup>-1</sup>
105		Not used
106		Not used
107	PTAUMU	Makeup flow time constant, s
108	PUAPM2	Steam-to-wall condensation heat transfer coeff × area, Btu/s-°F
109	PUAPS	Steam-to-liquid condensation heat transfer coeff × area, Btu/s-°F
110	MPW1	Initial mass in pressurizer lower liquid region, lb
111	MPW2	Initial mass in pressurizer upper liquid region, lb
112	HPW1	Initial enthalpy in pressurizer lower liquid region, Btu/lb
113	HPW2	Initial enthalpy in pressurizer upper liquid region, Btu/lb
114	MPS	Initial mass in pressurizer steam region, lb
115	HS	Initial enthalpy in pressurizer steam region, Btu/lb
116	MRC	Total mass in RC system (excluding pressurizer), lb
117	VRC	Total volume of RC system (excluding pressurizer), ft <sup>3</sup>
118	POWMIN	Initial power level after reactor trip, % of full power
119	DECAY	Decay heat level, % of full power
120	BOR	Reactivity insertion change due to boration or deboration, $\Delta k/k/h$
121	WMAX	Maximum secondary flow for analog scaling purposes, pps
122	DCA	Equivalent cross sectional flow area for downcomer, ft <sup>2</sup>
123	DCL	Total height of downcomer, ft
124	RASP	Friction and shock loss multiplier for steam aspiration port, psi/pps <sup>2</sup>
125	RORF	Friction and shock loss multiplier for inlet orifice, psi/pps <sup>2</sup>

Table 3-1. (Cont'd)

<u>No.</u>	<u>Name</u>	<u>Description</u>
126	RFDC	Feedwater inlet nozzle momentum multiplier, psi/pps <sup>2</sup>
127	DELT	Time step, s
128	ALPHA	Time scaling factor, computer time/problem time, dimensionless
129	PNOM	Nominal secondary pressure for analog scaling purposes, psia
130	PMAX	Maximum deviation (+ or -) from nominal secondary pressure, psi
131	TNOM	Nominal primary or secondary temperature for analog scaling purposes, F
132	TMAX	Maximum deviation (+ or -) from nominal temperature, F
133	HNOM	Nominal enthalpy of secondary fluid for analog scaling purposes, Btu/lbm
134	HMAX	Maximum deviation (+ or -) from nominal enthalpy, Btu/lbm
135	TAU6B	Time constant for HP turbine extraction flow to heater 6, s
136	TAU5B	Time constant for HP turbine extraction flow to heater 5, s
137	TAU4	Time constant for HP turbine extraction flow to heater 4, s
138	TAUGRH	Time constant for HP turbine extraction flow to reheater 1, s
139	TAU7	Time constant for HP turbine exist flow to moisture separator, s
140	TAU3	Time constant for LP turbine extraction flow to heater 3, s
141	TAU2	Time constant for LP turbine extraction flow to heater 2, s
142	TAU1	Time constant for LP turbine extraction flow to heater 1, s
143	TAU9	Time constant for LP turbine exit flow to condenser, s
144	TAURH1	Time constant for reheater 1 exit flow, s
145	TAURH2	Time constant for reheater 2 exit flow, s
146	TAUHP6	Time constant for HP feedwater heater 6, s
147	TAUHP5	Time constant for HP feedwater heater 5, s
148	TAUHP4	Time constant for HP feedwater heater 4, s
149	TAULP3	Time constant for LP feedwater heater 3, s
150	TAULP2	Time constant for LP feedwater heater 2, s

Table 3-1. (Cont'd)

<u>No.</u>	<u>Name</u>	<u>Description</u>
151	TAULP1	Time constant for LP feedwater heater 1, s
152	TAUDHP	Time constant for HP drain tank flow, s
153	TAUDLP	Time constant for LP drain tank flow, s
154	TAUFPT	Time constant for feedwater pump turbine flow, s
155	H6	Enthalpy of extraction flow 6, Btu/lbm
156	H6B	Enthalpy of extraction flow 6B, Btu/lbm
157	HPG6	Enthalpy of "pegging" steam (WPG6) to HP feedwater, heater 6
158	HRH2D	Enthalpy of reheater 2 drain flow (WRH2) Btu/lbm
159	H5	Enthalpy of steam to HP feedwater heater 5, Btu/lbm
160	H5B	Enthalpy of HP turbine extraction flow W5B, Btu/lbm
161	HPG5	Enthalpy of "pegging" steam (WPG5) to HP feedwater heater 5, Btu/lbm
162	H4	Enthalpy of HP turbine extraction flow W4, Btu/lbm
163	HRH1	Enthalpy of reheater 1 flow WRH1, Btu/lbm
164	HRH2	Enthalpy of reheater 2 flow WRH2, Btu/lbm
165	H8B	Enthalpy of feedwater pump turbine flow W8B, Btu/lbm
166	H3	Enthalpy of LP turbine extraction flow W3, Btu/lbm
167	H2	Enthalpy of LP turbine extraction flow W2, Btu/lbm
168	H1	Enthalpy of LP turbine extraction flow W1, Btu/lbm
169	DHFP	Enthalpy rise across main feedwater pumps, Btu/lbm
170	WPG6M	Maximum "pegging" steam to HP feedwater heater 6, pps
171	WPG5M	Maximum "pegging" steam to HP feedwater heater 5, pps
172	MFW1	Nominal liquid mass from feedwater heater 1 to 2, lbm
173	MFW2	Nominal liquid mass from feedwater heater 2 to 3, lbm
174	MFW3	Nominal liquid mass from feedwater heater 3 to 4, lbm
175	MFW4	Nominal liquid mass from feedwater heater 4 to 5, lbm
176	MFW5	Nominal liquid mass from feedwater heater 5 to 6, lbm
177	MFW	Nominal liquid mass from feedwater heater 6 to feedwater junction A-B, lbm
178	MFWA	Nominal liquid mass from feedwater junction A-B to OTSG-A, lbm
179	MFWB	Nominal liquid mass from feedwater junction A-B to OTSG-B, lbm

Table 3-1. (Cont'd)

<u>No.</u>	<u>Name</u>	<u>Description</u>
180	LAC1	Equivalent length/area ratio for condensate flow WC1, ft <sup>-1</sup>
181	LAC2	Equivalent length/area ratio for condensate flow WC2, ft <sup>-1</sup>
182	ARH0HW	Hotwell area times density, lbm/ft
183	PC	Pressure at condenser, psia
184	PC1	Pressure at junction of LP drain tank flow and heater 1, psia
185	PC2	Pressure at junction of HP drain tank flow and heater 4, psia
186	LHW	Hotwell level, ft
187	DPEC1	Elevation pressure differential between hotwell and PC2, psi
188	DPEC2	Elevation pressure differential between PC2 and PC1, psi
189	DPHWL	Initial pressure differential due to hotwell level, psi
190	HWLMIN	Minimum hotwell level allowed, ft
191	WC1	Condensate flow, condenser to junction at PC1, pps
192	WC2	Condensate flow, junction of PC1 to PC2, pps
193	DELTNO	Number of time steps between updates of secondary system temperature delays (dimensionless)
194	DPSGMX	Maximum steam generator secondary pressure drop (psi) for analog scaling purposes
195	TFW0FF	Arbitrary feedwater temperature offset, F
196- 200	DDUM(1) DDUM(5)	Dummy variables, no used

Table 3-2. Tabular Data

No.	Function	Description
1	$\%R_r = F_r(\%X_r)$	Control rod worth (%) Vs insertion (%)
2	$\%R_{sr} = F_{sr}(t)$	Safety rod worth (%) Vs time after trip (s)
3	$W_{pa} = F_{1a}(t)$	One-primary-pump trip, loop A flow rate (pps) Vs time after trip (s)
4	$W_{pb} = F_{1b}(t)$	One-primary-pump trip, loop B flow rate (pps) Vs time after trip (s)
5	$W_{pa} = F_{2a}(t)$	Four-primary-pump trip, loop A flow rate (pps) Vs time after trip (s)
6	$W_{pb} = F_{2b}(t)$	Four-primary-pump trip, loop B flow rate (pps) Vs time after trip (s)
7	$W_{pa} = F_{3a}(t)$	Two-primary-pump trip in same loop, loop A flow rate (pps) Vs time after trip (s)
8	$W_{pb} = F_{3b}(t)$	Two-primary-pump trip in same loop, loop B flow rate (pps) Vs time after trip (s)
9	$W_{pa} = F_{4a}(t)$	Two-primary-pump trip, 1/loop, loop A flow rate (pps) Vs time after trip (s)
10	$W_{pb} = F_{4b}(t)$	Two-primary-pump trip, 1/loop, loop B flow rate (pps) Vs time after trip (s)
11	$W_{6b} = F_1(W_t)$	Extraction flow $W_{6b}$ (pps) to HP heater 6 Vs turbine throttle flow $W_t$ (pps)
12	$W_{5b} = F_2(W_t)$	Extraction flow $W_{5b}$ (pps) to HP heater 5 Vs turbine throttle flow $W_t$ (pps)
13	$W_4 = F_3(W_t)$	Extraction flow $W_4$ (pps) to HP heater 4 Vs turbine throttle flow $W_t$ (pps)
14	$h_{hp6} = F_4(W_t)$	Enthalpy $h_{hp6}$ (Btu/lb) of FW leaving HP heater 6 Vs turbine throttle flow $W_t$ (pps)
15	$h_{hp5} = F_5(W_t)$	Enthalpy $h_{hp5}$ (Btu/lb) of FW leaving HP heater 5 Vs turbine throttle flow $W_t$ (pps)
16	$W_{ms} = F_1(W_7)$	Moisture separator flow $W_{ms}$ to high pressure drain tank (pps) Vs flow $W_7$ (pps) out of HP turbine
17	$M_{s7} = F_2(W_7)$	Torque developed $M_{s7}$ (%) by HP turbine Vs steam flow $W_7$ (pps) out of HP turbine

Table 3-2. (Cont'd)

No.	Function	Description
18	$h_{ms} = F_3(W_7)$	Enthalpy (Btu/lb) of moisture separator flow $W_{ms}$ Vs flow $W_7$ (pps) out of HP turbine
19	$W_3 = F_1(W_8)$	Extraction flow $W_3$ (pps) out of LP turbine Vs flow $W_8$ (pps) into LP turbine
20	$W_2 = F_2(W_8)$	Extraction flow $W_2$ (pps) out of LP turbine Vs flow $W_8$ (pps) into LP turbine
21	$W_1 = F_3(W_8)$	Extraction flow $W_1$ (pps) out of LP turbine Vs flow $W_8$ (pps) into LP turbine
22	$W_{rh2} = F_1(W_{8c})$	Live steam extraction flow $W_{rh2}$ (pps) to reheater 2 Vs flow $W_{8c}$ (pps) through reheater
23	$W_{8b} = F_2(W_{8c})$	Feedwater pump turbine flow $W_{8b}$ (pps) Vs flow $W_{8c}$ (pps) through reheater 2
24	$W_{rh1} = F_6(W_t)$	Extraction flow $W_{rh1}$ (pps) from HP turbine to reheater 1 Vs turbine throttle flow $W_t$ (pps)
25	$h_{rh1d} = F_1(W_{8d})$	Enthalpy $h_{rh1d}$ (Btu/lb) of reheater 1 drain flow $W_{rh1}$ (pps) Vs flow $W_{8d}$ (pps) through reheater 1
26	$M_{89} = F_1(W_9)$	Torque developed $M_{89}$ (%) by LP turbine Vs steam flow $W_9$ (pps) out of LP turbine
27	$h_{6d} = F_1(W_{fw})$	Enthalpy $h_{6d}$ (Btu/lb) of HP feedwater heater 6 drain flow $W_{6d}$ (pps) Vs feedwater flow $W_{fw}$ (pps)
28	$h_{5d} = F_2(W_{fw})$	Enthalpy $h_{5d}$ (Btu/lb) of HP feedwater heater 5 drain flow $W_{6d}$ (pps) Vs feedwater flow $W_{fw}$ (pps)
29	$h_{4d} = F_1(W_{c2})$	Enthalpy $h_{4d}$ (Btu/lb) of HP feedwater heater 4 drain flow $W_{4d}$ (pps) Vs condensate flow $W_{c2}$ (pps)
30	$h_{3d} = F_2(W_{c2})$	Enthalpy $h_{3d}$ (Btu/lb) of LP feedwater heater 3 drain flow $W_{3d}$ (pps) Vs condensate flow $W_{c2}$ (pps)
31	$h_{2d} = F_3(W_{c2})$	Enthalpy $h_{2d}$ (Btu/lb) of LP feedwater heater 2 drain flow $W_{2d}$ (pps) Vs condensate flow $W_{c2}$ (pps)
32	$h_{1d} = F_1(W_{c1})$	Enthalpy $h_{1d}$ (Btu/lb) of LP feedwater heater 1 drain flow $W_{1d}$ (pps) Vs condensate flow $W_{c1}$ (pps)
33	$\Delta P_{hwp} = f_2(W_{c1})$	Hotwell pump developed head (psi) Vs flow $W_{c1}$ (pps) through pump
34	$\Delta P_{cp} = F_3(W_{c1})$	Condensate pump developed head (psi) Vs flow $W_{c1}$ (pps) through pump

Table 3-2. (Cont'd)

<u>No.</u>	<u>Function</u>	<u>Description</u>
35	$h_c = F_4(W_{c1})$	Enthalpy $h_c$ (Btu/lb) at LP heater 1 inlet Vs flow $W_{c1}$ (pps) through heater
36	$h_{fpt} = F_7(W_t)$	Enthalpy $h_{fpt}$ (Btu/lb) of feedwater pump turbine flow $W_{fpt}$ (pps) Vs turbine throttle flow $W_t$ (pps)

Table 3-3. COMANCHE Parameter Data

ALPHA	Time scale factor, computer time/problem time
ALPRIP	Reciprocal of ALPHA
AM	Steam generator metal cross-sectional area, ft <sup>2</sup>
AP	Steam generator primary cross-sectional area, ft <sup>2</sup>
AS	Steam generator secondary cross-sectional area, ft <sup>2</sup>
B	Drift velocity multiplier
BETA	Spatial integration rate of steam generator, ft/s
COND	Tube metal conductivity × 100, Btu/s-F-ft
CPM	Specific heat of steam generator metal, Btu/lb-F
CPW	Specific heat of primary fluid, Btu/lb-F
CVA	Percent feed valve A coefficient at WFWA and XCVA
CVB	Percent feed valve B coefficient at WFWB and XCVB
CVSF	Scale factor (chosen to make DP = 1000 when A33, A38 are limited)
CO	Void fraction distribution coefficient
C5P1	Drift velocity times R8/B, at P1 pressure
C5P2	Drift velocity times R4/B, at P2 pressure
C5P3	Drift velocity times R9/B, at P3 pressure
DNBX	Steam quality at DNB
DPCVA	Feed valve A pressure drop at WFWA and XCVA, psia
DPCVB	Feed valve B pressure drop at WFWB and XCVB, psi
DPCVMX	Max feed valve pressure drop, psi
DPEFM	Elevation pressure gain, main feed line, psi
DPEFWA	Elevation pressure drop, feed line A, psi
DPEFWB	Elevation pressure drop, feed line B, psi

Table 3-3. (Cont'd)

DPFA	Friction pressure drop, feed line A, psi
DPFB	Friction pressure drop, feed line B, psi
DPFW	Friction pressure drop, main feed line, psi
DPSFSC	Subcooled region unrecovered delta-P, psi/ft (at WFW)
DPSFBC	Boiling region unrecovered delta-P, psi/ft (at WFW)
DPSFSH	Superheat region unrecovered delta-P, psi/ft (at WFW)
DPSMAX	Maximum SG pressure drop, psi
DPUMPR	Main feed pump rated heat at rated flow and speed, $D(\rho)/DP$ of steam at constant enthalpy (1234.7)
DT	Major cycle time of hybrid steam generator-model
EBPLOT	Delay time for emergency feed system, s
ECV	Control valve position error for maximum velocity, %
EXP1	EXP(-P1/1260) for heat flux calculation
EXP2	EXP(-P2/1260) for heat flux calculation
EXP3	EXP(-P3/1260) for heat flux calculation
FW1	Minimum feedwater temperature, F
FW2	Normal feedwater temperature, F
FW3	Maximum feedwater temperature, F
F1	Saturated fluid enthalpy at P1 pressure, Btu/lb
F2	Saturated fluid enthalpy at P2 pressure, Btu/lb
F3	Saturated fluid enthalpy at P3 pressure, Btu/lb
G1	Saturated gas enthalpy at P1 pressure, Btu/lb
G2	Saturated gas enthalpy at P2 pressure, Btu/lb
G3	Saturated gas enthalpy at P3 pressure, Btu/lb
HFW	Feedwater enthalpy, Btu/lbm
HMAX	Scaling parameter for secondary enthalpy (range), Btu/lb
HNOM	Scaling parameter for secondary enthalpy (midpoint), Btu/lb
H1	Subcooled enthalpy at FW1 and P2, Btu/lb
H2	Subcooled enthalpy at FW2 and P2, Btu/lb
H4	Superheated enthalpy at T4 and P2, Btu/lb
L	Active length of steam generator, ft
LAFW	Length to area ratio, main feed line, 1/ft
LAFWA	Length to area ratio, feed line A, 1/ft
LAFWB	Length to area ratio, feed line B, 1/ft
LA1A	Steam line 1A length to area ratio, 1/ft

Table 3-3. (Cont'd)

LA1B	Steam line 1B length to area ratio, 1/ft
LA2A	Steam line 2A length to area ratio, 1/ft
LA2B	Steam line 2B length to area ratio, 1/ft
LA3A	Steam line 3A length to area ratio, 1/ft
LA3B	Steam line 3B length to area ratio, 1/ft
LDPSAV	Level in SG where total DP is sampled and saved, ft
MMAX	Maximum steam dump to atmosphere per SG
NPUMPF	Main feed pump final speed after trip, %
NPUMPI	Main feed pump initial speed, %
NPJMPR	Main feed pump rated speed, %
PBP	Bypass valve pressure at rated flow (WBP), psia
PC2	Normal feedpump suction pressure, psia
PC2REF	Feed pump suction pressure at which head was measured
PDMAX	Scaling parameter for PSL2A-PT and PSL2B-PT, psi
PESF	Low-pressure setpoint for containment isolation, psia
PMAD	Main atmospheric dump valve rated pressure, psia
PMAX	Scaling parameter for secondary pressure (range and midpoint), psia
PNPSH	Main feed pump trip setpoint on NPSH, psia
POWER	Neutron power, %
PP	Primary wetted perimeter, ft
PRC	Primary system pressure, psia, in pressurizer steam space
PS	Secondary wetted perimeter, ft
PSAD	Small atmospheric dump valve pressure, psia
PSGOA	Steam generator A outlet pressure, psia
PSGOB	Steam generator B outlet pressure, psia
PSL1A	Pressure at steam line node 1A, psia
PSL1B	Pressure at steam line node 1B, psia
PSL2A	Pressure at main atmospheric dump valve A, psia
PSL2B	Pressure at main atmospheric dump valve B, psia
PSV	Steam line safety valve rating pressure (103% accumulation), psia
PT	Pressure at turbine valve, psia
P1	Minimum expected pressure, psia
P2	Expected normal pressure, psia
P3	Maximum expected pressure, psia

Table 3-3. (Cont'd)

RESET1	Reset pressure of lowest bank of steam line safety valve, s
RM	Steam generator metal density, lb/ft <sup>3</sup>
RP	Steam generator primary density, lb/ft <sup>3</sup>
RSH1	Steam density at P1 and enthalpy at P2 and T4
RSH3	Steam density at P3 and enthalpy at P2 and T4
R1	Subcooled density at FW1 and P2, lb/ft <sup>3</sup>
R2	Saturated fluid density at T2 and P2, lb/ft <sup>3</sup>
R3	Saturated fluid density at T3 and P3, lb/ft <sup>3</sup>
R4	Saturated gas density at T2 and P2, lb/ft <sup>3</sup>
R5	Superheated density at T4 and P2, lb/ft <sup>3</sup>
R6	Superheated density at T4 and P1, lb/ft <sup>3</sup>
R7	Saturated fluid density at T1 and P1, lb/ft <sup>3</sup>
R8	Saturated gas density at T1 and P1, lb/ft <sup>3</sup>
R9	Saturated gas density at T3 and P3, lb/ft <sup>3</sup>
SGCOMP	=1 if SG component option is used (=0 otherwise)
STROKE	Feedwater control valve stroke time to open, s
TAUCV	Feed valve motor time constant, s
TAUSV	Main steam safety valve opening and closing time constant, s
TEMPCL	Cold leg temperature, F
TEMPHL	Hot leg temperature, F
TMAX	Scaling parameter for secondary temperature (range), F
TNOM	Scaling parameter for secondary temperature (midpoint), F
TFWI	FW line isolation valve closing time, s
TSLI	Steam line isolation valve closing time, s
T1	Saturation temperature at P1 pressure, F
T2	Saturation temperature at P2 pressure, F
T3	Saturation temperature at P3 pressure, F
T4	Expected full power steam temperature, F
UPM	Primary-metal heat transfer coefficient, Btu/s-ft <sup>2</sup> -°F
USC	Secondary subcooled heat transfer coefficient (at WFW), Btu/s-ft <sup>2</sup> -°F
UNBMUL	MUL on Thom's coefficient
USH	Secondary superheat heat transfer coefficient (at WFW), Btu/s-ft <sup>2</sup> -°F
V1A	Steam line 1A volume, ft <sup>3</sup>
V1B	Steam line 1B volume, ft <sup>3</sup>

Table 3-3. (Cont'd)

V2A	Steam line 2A volume, ft <sup>3</sup>
V2B	Steam line 2B volume, ft <sup>3</sup>
V3A	Steam line 3A volume, ft <sup>3</sup>
V3B	Steam line 3B volume, ft <sup>3</sup>
VT	Volume of turbine line from bypass junction, ft <sup>3</sup>
WBP	Total condenser dump at 1050 psia (18% of 2*WFW), lb/s
WEFWA	Emergency feedwater flow, loop A maximum, lb/s
WEFWB	Emergency feedwater flow, loop B maximum, lb/s
WFW	Maximum feedwater flow into steam generator, lb/s
WMAD	Main atmospheric dump valve rated flow at PMAD (per OTSG), lb/s
WMAX	Maximum feedwater flow into steam generator, lb/s
WP	Actual primary flow through steam generator, lb/s
WPMAX	Maximum primary flow rate, lb/s
WRATED	Main feed pump rated flow per pump, lb/s
WSAD	Small atmospheric dump flow at PSAD and XSAD=100 psi
WSV	Steam line safety valve flow (112% of WFW), lb/s
WT	Turbine throttle flow at 100% power
XCVA	Feed valve A initial position, %
XCVB	Feed valve B initial position, %
XC5	Quality for start of ramp C5 to zero
XTV	Turbine valve initial position, %
ZERO	
MS	Full power steam torque, %
FREQ	Normal grid frequency, Hz
FREQDN	Negative change in grid frequency, Hz
FREQUP	Positive change in grid frequency, Hz
KTG1	T-G speed change rate, 50 Hz/9.95 s
KTG2	Normal house load, %
KTG3	T-G torque change per load angle change, 360/64.29
KTG4	T-G damping constant, percent per delta frequency
KTG5	T-G friction and windage torque per Hz, %
MWE	Normal MW electric net to grid, %

Table 3-4. Optional Output Variables

<u>Output No.</u>	<u>Variable name</u>	<u>Description</u>
1	WPA	Loop A RC flow, lb/s
2	WPB	Loop B RC flow, lb/s
3	MPW1	Mass in pressurizer water region 1, lb
4	MPW2	Mass in pressurizer water region 2, lb
5	HPW1	Enthalpy of pressurizer water region 1, Btu/lb
6	HPW2	Enthalpy of pressurizer water region 2, Btu/lb
7	MPS	Mass in pressurizer steam space, Btu/lb
8	HS	Enthalpy of pressurizer steam space, Btu/lb
9	PC1	Pressure of feedwater at feed heater outlet 1, psia
10	PC2	Pressure of feedwater at MFP suction, psia
11	LHW	Water level in condenser hotwell, ft
12	WC1	Feedwater flow at outlet of feed heater 1, lb/s
13	WC2	Feedwater flow upstream of MFP suction, lb/s
14	TAVEM	$(TC+TH)/2$ at the output of 5 s time constant, F
15	TCA	Loop A RC temperature at SG outlet, F
16	TCB	Same for loop B, F
17	THA	Loop A RC temperature at SG inlet, F
18	THB	Same for loop B, F
19	TRIA	Loop A RC temperature at reactor vessel inlet, F
20	TRIB	Same for loop B, F
21	TRO	RC temperature at reactor core outlet, F
22	TROA	RC temperature at reactor vessel loop A outlet, F
23	TROB	Same for loop B, F
24	TSGIA	Loop A RC temperature at SG tubesheet inlet, F
25	TSGIB	Same for loop B, F
26	TWI	RC temperature at reactor core inlet, F
27	TW	RC temperature at reactor core outlet, F
28	POWER	Reactor core NI power, %
29	PTHERM	Reactor core heat flux at clad surface, %
30	TF	Reactor core fuel temperature, F
31	TC	Reactor core clad temperature, F
32	TW	Reactor core average water temperature, F

Table 3-4. (Cont'd)

<u>Output No.</u>	<u>Variable name</u>	<u>Description</u>
33	XR	Reactor core control rod position, %
34	DELKT	Reactor core total reactivity, $\Delta k/k$
35	TSGOA	Loop A RC temperature at SG tubesheet outlet, F
36	TSGOB	Same for loop B, F
37	WP	Total RC flow, lb/s
38	WMU	RC makeup flow, lb/s
39	WSP	Pressurizer spray flow, lb/s
40	WR	Pressurizer safety valve flow, lb/s
41	WPR	Pressurizer power actuated relief valve flow, lb/s
42	LP	Pressurizer water level, in.
43	TPW1	Pressurizer water region 1 temp, F
44	TPW2	Pressurizer water region 2 temp, F
45	ROPS	Pressurizer steam space density, lb/ft <sup>3</sup>
46	TPS	Pressurizer steam space temp, F
47	WC4	Pressurizer spray condensation rate, lb/s
48	QHTR	Pressurizer total heat output to water, Btu/s
49	PP	RC pressure in pressurizer steam space, psia
50	PRC	RC pressure at core outlet, psia
51	WB	Pressurizer vaporization rate, lb/s
52	WC5	Pressurizer interface condensation rate, lb/s
53	WC6	Pressurizer wall condensation rate, lb/s
54	WSU	RC flow through surge line, lb/s
55	DCLENA	Loop A SG downcomer length, ft
56	DCLENB	Loop B SG downcomer length, ft
57	WFWA	Feedwater flow, loop A, lb/s
58	WFWB	Feedwater flow, loop B, lb/s
59	WASPA	Loop A SG aspiration flow rate, lb/s
60	WASPB	Loop B SG aspiration flow rate, lb/s
61	WFA	SG tubesheet inlet flow, loop A, lb/s
62	WFB	SG tubesheet inlet flow, loop B, lb/s
63	DPOTA	Pressure differential between 0 and 32 ft, SG-loop A, psi
64	DPOTB	Pressure differential between 0 and 32 ft, SG-loop B, psi

Table 3-4. (Cont'd)

<u>Output No.</u>	<u>Variable name</u>	<u>Description</u>
65	HFWA	Enthalpy of loop A feedwater, Btu/lb
66	HFWB	Enthalpy of loop B feedwater, Btu/lb
67	PSGOA	SG outlet pressure, loop A, psia
68	PSGOB	SG outlet pressure, loop B, psia
69		Not used
70		Not used
71	HASPA	Enthalpy of aspiration steam, SG-A, Btu/lb
72	HASPB	Enthalpy of aspiration steam, SG-B, Btu/lb
73	TSCA	Average RC temperature, SG-A, F
74	TSGB	Average RC temperature, SG-B, F
75	WSGOA	Steam flow, SG-A outlet, lb/s
76	WSGOB	Steam flow, SG-B outlet, lb/s
77	TSTMA	Steam temperature, SG-A outlet, F
78	TSTMB	Steam temperature, SG-B outlet, F
79	LSCA	Subcooled length, SG-A, ft
80	LSCB	Subcooled length, SG-B, ft
81	LNBA	Nucleate boiling length, SG-A, ft
82	LNBB	Nucleate boiling length, SG-B, ft
83	QSGA	Primary-to-secondary heat transfer, SG-A, %
84	QSCB	Primary-to-secondary heat transfer, SG-B, %
85	W1	Turbine extraction flow, lb/s
86	W1D	Heater drain flow, lb/s
87	W2	Extraction flow, lb/s
88	W2D	Heater drain flow, lb/s
89	W3	Extraction flow, lb/s
90	W3D	Heater drain flow, lb/s
91	W4	Extraction flow, lb/s
92	W4D	Heater drain flow, lb/s
93	W5	Extraction flow, lb/s
94	W5D	Heater drain flow, lb/s
95	W5B	Extraction flow, lb/s
96	W6	Extraction flow, lb/s
97	W6D	Heater drain flow, lb/s

Table 3-4. (Cont'd)

<u>Output No.</u>	<u>Variable name</u>	
98	W6B	Extraction flow, lb/s
99	W7	HP turbine outlet flow, lb/s
100	W8	LP turbine inlet flow, lb/s
101	W8A	LP turbine inlet valve flow, lb/s
102	W8B	Feed pump extraction flow, lb/s
103	W8C	Reheater 2 outlet flow, lb/s
104	W8D	Reheater 1 outlet flow, lb/s
105	W9	LP turbine outlet flow, lb/s
106	WMS	Moisture separator drain flow, lb/s
107	WRH1	Extraction flow to reheater 1, lb/s
108	WRH2	Reheater 2 drain flow, lb/s
109	WPG5	Pegging flow to feed heater 5, lb/s
110	WPG6	Pegging flow to feed heater 6
111	WDLF	Flow out of LP drain tank, lb/s
112	WDHP	Flow out of HP drain tank, lb/s
113	WFPT	Flow through feed pump turbine, lb/s
114	MS7	HP turbine steam torque, %
115	MS9	LP turbine steam torque, %
116	MS	Total turbine steam torque, %
117	HC1	Feedwater enthalpy 1 heater exit, Btu/lb
118	HC2	Feedwater enthalpy MFP suction, Btu/lb
119	HLP1	Feedwater enthalpy 1 heater exit, Btu/lb
120	HLP2	Feedwater enthalpy 2 heater exit, Btu/lb
121	HLP3	Feedwater enthalpy 3 heater exit, Btu/lb
122	HHP4	Feedwater enthalpy 4 heater exit, Btu/lb
123	HHP5	Feedwater enthalpy 5 heater exit, Btu/lb
124	HHP6	Feedwater enthalpy 6 heater exit, Btu/lb
125	HDLF	Feedwater enthalpy LP drain tank exit, Btu/lb
126	HDHP	Feedwater enthalpy HP drain tank exit, Btu/lb
127	H1D	Drain enthalpy, heater 1, Btu/lb
128	H2D	Drain enthalpy, heater 2, Btu/lb
129	H3D	Drain enthalpy, heater 3, Btu/lb
130	H4D	Drain enthalpy, heater 4, Btu/lb

Table 3-4. (Cont'd)

<u>Output No.</u>	<u>Variable name</u>	<u>Description</u>
131	H5D	Drain enthalpy, heater 5, Btu/lb
132	H6D	Drain enthalpy, heater 6, Btu/lb
133	HMS	Moisture separator enthalpy, Btu/lb
134	HRH1D	Reheater 1 drain enthalpy, Btu/lb
135	DPHWP	Developed head, hotwell pump, psia
136	DPCP	Developed head, condensate pump, psia
137	DPFC1	Pressure losses, condenser to heater 1 exit, Btu/lb
138	DPFC2	Pressure losses, exit 1 to MFP suction, Btu/lb
139	TFW	Feedwater temperature, heater exit 6, F
140	TFWA	Feedwater temperature, SG-A inlet, F
141	TFWB	Feedwater temperature, SG-B inlet, F
142	WFW	Feedwater flow (total, A+B), lb/s
143	WT	Turbine inlet flow, lb/s

Table 3-5. Commonly Displayed Analog Variables

ICS outputs (such as ULD)

Secondary plant variables

Throttle pressure

Steam line safety valve flow

Bypass valve flow

Atmospheric dump flow

Feed pump speed or head

Feedwater control valve position or pressure drop

Megawatts electric

Throttle valve position

## 4. POWER TRAIN APPLICATIONS

### 4.1. Applications

#### 4.1.1. General

POWER TRAIN models the B&W two-loop nuclear steam system (NSS) including two different types of once-through steam generators (OTSGs); it simulates the NSS and the major parts of the balance-of-plant (BOP) systems that are involved in the power generation process. These major systems and components include the following:

#### Reactor Coolant System (RCS)

- Reactor vessel and core
- RC pumps
- Pressurizer
- RC pressure control
  - Heaters
  - Spray
  - PORV
  - Pressurizer safety valves
- Control rods and drives
  - Control groups
  - Safety groups
- Connecting piping
  - Hot leg
  - Cold leg
  - Surge line

#### Steam System

- Steam lines
- Steam safety valves
- Turbine bypass system
  - Condenser dump
  - Atmospheric vent
- Modulating atmospheric dump valves
- Turbine
  - Turbine control/stop valves

- Generator
- Condenser
- Reheaters/moisture separators
- Appropriate ICS controls

#### Condensate System

- Hot well
- Hot well pumps
- Condensate booster pumps
- Connecting piping
- Flash tanks

#### Feedwater System

- Deaerating feedwater heater
- Main feed pumps
- Feedwater control valves
- Appropriate ICS controls

The equations and mathematical relations for these systems and components are provided in section 2.

The techniques described here are applicable to B&W 145-, 177-, and 205-fuel assembly NSSs and corresponding BOP equipment. While the modeling techniques are consistent for all plants, the details of the simulation will vary primarily with the configuration of the BOP hardware. The major modeling variations from plant to plant involve the condensate and feedwater systems, deaerating feedwater heaters, condensate booster pumps, and constant-speed main feedwater pumps. These variations depend on utility/AE practice or preference. The RCS simulation configuration is generally fixed, and the variations from one NSS to another are handled by variations in the input data consistent with the specific unit being modeled.

The POWER TRAIN model described here contains some 55 individual components, which are shown in modular form in Figure 4-1. The model also contains the integrated control system (ICS), the major functions of the reactor protection system (RPS), and portions of the engineered safety features actuation system (ESFAS).

This section, along with section 2, provides (1) application details of the POWER TRAIN simulation, (2) descriptions of the ICS, RPS, and ESFAS, and (3) typical results and sensitivity studies. The appendix compares this POWER TRAIN model version with an SAR Chapter 15 event analyzed using an earlier version.

#### 4.1.2. Design, Performance, and Control

Because of its ability to execute transients in real time, POWER TRAIN is used for a wide range of performance and control applications. The following major activities are included in performance and control:

##### Design

- Development of control system design requirements
- Verification of control system design
- Development of control system gain, integration ratio, setpoints, and derivative rates
- Development of valve characteristics including capacity, setpoints, stroke speed, and flow-versus-stroke characteristics
- Pressurizer volume determination/verification
- Control rod speed/reactivity insertion rates

##### Performance

- Development of load change requirements
- Verification of load change capability
- Setpoint verification
- Determination of operability and operational margins
- Development of transient response characteristics for input to stress analysis

##### Control

- Development of control system design requirements
- Verification of control system design
- Development of control system gain, integration rates, differential rates, and setpoints
- Analysis tool as part of failures modes and effects analysis (FMEA)

#### 4.1.3. Safety Analysis Application

POWER TRAIN is used to analyze or to provide input to other codes for up to 10 different transients for FSAR Chapter 15 analyses. When the code is used to provide input to other codes, its output consists of steady-state and transient steam generator heat demand responses during accident transients. The transient heat demand response is input to the digital code CADD, which in turn predicts the primary system responses for FSAR application. The CADD results are then input to RADAR for transient fuel temperatures, cladding temperatures, and departure from nucleate boiling ratio (DNBR) results when applicable. POWER TRAIN is used to provide steam generator responses for the following transients:

1. Decrease in feedwater temperature
2. Increase in feedwater flow
3. Increase in steam flow
4. Decrease in steam flow
5. Loss of external load
6. Loss of onsite and offsite power
7. Inadvertent closing of main steam isolation valve
8. Loss of all feedwater
9. RC pump trips
10. Loss of condenser vacuum

For normal transient analyses, such as turbine trip or load rejection, (assuming nominal conditions) POWER TRAIN output is used for SAR Chapter 15 applications. Figure 4-2 is an example of POWER TRAIN results for a turbine trip with power runback to 15% power.

POWER TRAIN is also used for the overpressure protection analysis in SAR Chapter 5 (5.2.2).

Finally, the program is used to provide input for normal load change data for SAR Chapter 4 DNB analyses.

#### 4.2. Inputs to POWER TRAIN

The program requires certain inputs from other sources. The major inputs and their sources are as follows:

1. Reactor coolant flow characteristics:
  - System flow rates, flow coastdown, and pump startup characteristics from the PUMP and SPLIT digital codes
  - System pressure drops from the SAVER code
2. Steam and feedwater systems parameters
  - Flow rates and valve characteristics from design calculations and manufacturers' data
3. Steam generator model:
  - Heat transfer characteristics from Alliance Research Center (ARC) test data and from digital codes PIEOTSG (145- and 205-FA NSS) and POTSG (177-FA NSS).
4. Core physics characteristics:
  - Decay heat from  $\text{ANSI } 5.1 \times 1.0$  (or appropriate multiplier)
  - Rod worth from physics calculation codes (PDQ and FLAME)
  - Doppler/moderator coefficients from physics calculation codes (PDQ and FLAME)
  - Cladding and fuel temperatures from thermal-hydraulic calculation code (CHATA)
5. Component volumes:
  - Primary system components from B&W design documentation and drawings
  - Secondary system: steam, feedwater and condensate systems
  - Turbine and condenser — from customer design documentation and drawings
6. Secondary side parameters:
  - Design data from customer and/or customer vendors' design documentation and drawings

#### 4.3. Modeling Representations in POWER TRAIN

##### 4.3.1. Valves

Ten different types of valves are represented in POWER TRAIN, each of which has its own characteristic response. The valves and their modeling characteristics are described below.

<u>Valve description</u>	<u>No. of valves</u>	<u>Model characteristics</u>
Main feedwater	2 (one/loop)	Full-range control of feedwater flow. Valve opens and closes at linear rates. Valve flow coefficients Vs stroke is modeled to actual valve characteristics.
Turbine throttle/ turbine stop	1	One valve simulates full-range steam flow corresponding to full-arc turbine admission. Valve strokes linearly per turbine vendor design. Flow coefficient Vs stroke is adjustable to specific turbine date. Valve is also modeled to close rapidly to simulate turbine stop valve.
Feedwater isolation	2 (one/loop)	Opens or closes within 0.25 second on command.
Main steam safety	4 (two/loop)	Open or close at adjustable pressure setpoints. Valves stroke full-open or full-closed with response of 0.25-second time constant.
Steam line isolation	2 (one/loop)	Open or close within 0.25 second on ESFAS command.
Condenser steam bypass	1	One valve represents turbine bypass up to 25% steam flow to the condenser. The valve modulates on ICS control signal. Stroke response and flow coefficient are modeled to represent specific valves.
Modulating atmospheric vents	2 (one/loop)	Same general characteristics as above, except those valves vent steam to the atmosphere.
Main atmospheric vents (on-off)	8 (four/loop)	Valves vent steam to the atmosphere. They open and close to full capacity with response through 0.5-second time constant.
Pressurizer PORV	1	Valve ramps open or closed at adjustable rates based on valve characteristics. Valve flow coefficient based on manufacturer's data.
Pressurizer spray	1	Ramps open or close at adjustable constant velocity.
Pressurizer safety	1	Open and close instantaneously on adjustable pressure setpoints.

#### 4.3.2. Pumps

POWER TRAIN models two reactor coolant (RC) pumps (one in each loop), one main feedwater pump, one hot well pump, and one condenser pump (pumps shown in Figure 4-1). The RC pump model consists of tabular input of the RC loop flow rates for one-, two-, three-, and four-pump operation. The characteristics of these flow rates are obtained from the digital codes SPLIT and PUMP and from pump design data. The feedwater pump model is a homogeneous relationship of pump speed, flow rate, and pressure. The hot well and condenser pumps are modeled in POWER TRAIN by a head versus capacity curve.

#### 4.3.3. Integrated Control System Model

The ICS is the main control system in the B&W NSS. Its purpose is to automatically coordinate the (1) reactor power, (2) feedwater flow, and (3) turbine steam flow/bypass flow. Figure 4-3 is a functional diagram of the ICS modeled in POWER TRAIN. This model includes all the pertinent ICS functions to adequately simulate ICS action during application transients.

The POWER TRAIN model contains the following ICS controls and control subsystems:

<u>Subsystem</u>	<u>Control</u>
Unit master	Sets unit load demand for steady-state operation and the limits for asymmetric operation (i.e., three RC pumps). It also sets the various ramp rates for operational transients.
Integrated master	Controls the turbine throttle valve based on unit load demand (ULD) and provides setpoints for the turbine bypass system.
Reactor control	Positions control rods based on ULD and $T_{avg}$ .
Feedwater control	Positions the feedwater control valves and feedwater pump speed for a given ULD.

For safety analysis transients, the ICS is assumed operable, inoperable, or partially inoperable (depending on which is more conservative). For operational transients the ICS will usually be fully operable in order to predict expected plant conditions.

#### 4.3.4. Reactor Protection System

The POWER TRAIN model contains the major functions of the reactor protection system (RPS):

- High RC pressure
- Low RC pressure
- High neutron flux
- Maximum allowable ratio of neutron flux to RC flow
- High pressurizer level
- Low pressurizer level

Figure 4-4 shows the general implementation of the RPS trips in POWER TRAIN.

#### 4.3.5. Engineered Safety Features Actuation System (ESFAS)

POWER TRAIN simulates a partial ESFAS system. Figure 4-5 shows the POWER TRAIN ESFAS implementation. This ESFAS model includes the isolation of main steam and main feedwater when pressure in one or both steam generators decreases to a low setpoint. Forty seconds after the ESFAS signal, emergency feedwater is initiated.

#### 4.3.6. Sensors

PT models sensors that control or initiate ICS, RPS, and ESFAS action:

1. Reactor cold leg temperature in both loops (°F)
2. Reactor hot leg temperature in two loops (°F)
3. Reactor coolant pressure (psig)
4. Feedwater temperature (°F)
5. Steam generator pressure in both loops (psig)
6. Turbine header pressure (psig)
7. RC flow rate in both loops (lb/s)
8. Pressurizer level (in.)
9. Neutron flux (%)
10. Steam generator level (in. H<sub>2</sub>O)

The calculated pressure signals and neutron flux initiate a desired action instantaneously. The temperature signals used for control or prediction are processed through time constants and/or delays to initiate an action. The level signals (pressurizer and steam generator) initiate action instantaneously after a setpoint is reached.

#### 4.4. Application Results

POWER TRAIN is used to predict NSS transient responses for a wide variety of events. Selected transients from the 177- and 205-FA NSS models are included here to illustrate the code's response:

1. Loss of one RC pump from 90% power
2. Turbine trip from full power
3. Reactor trip from full power

##### 4.4.1. Loss of One RC Pump From 90% Power

In addition to the normal controlled transient description, this transient is described with and without ICS action to illustrate typical analyses.

##### 4.4.1.1. With ICS Control - 177-FA NSS

The ICS establishes a maximum allowable power level based on the number of RC pumps operating; for three RC pumps the limit is 75% power. The ICS continually compares the demand power level to the maximum allowable power. For this transient, the power is assumed to be 90% initially, and the maximum allowable is 75% power due to the loss of one RC pump (see S1 on Figure 4-3). This mismatch initiates a 50% per minute power demand runback (see A1 on Figure 4-3), which decreases the core power and feedwater flow at the maximum possible rate (~30% per minute) by ramping the feedwater valves closed and by inserting control rods at a maximum rate of 30 inches/minute. When the power demand is equal to or less than 75%, the runback terminates.

Upon trip of one RC pump, the ICS distributes feedwater flow to each steam generator based on the change in RC loop flows (see D10 on Figure 4-3). This ICS action keeps the  $\Delta T$  across both steam generators balanced and thereby maintains equal reactor vessel inlet temperatures.

After a runback period of about 30 seconds, the ULD steadies out at about 75% power. The actual reactor power may differ from the ULD ramp rate until the core  $T_{avg}$  is satisfied ( $\Delta_{11}$  on Figure 4-3). The transient terminates in about 2 minutes. Figure 4-6 shows the response predicted for this transient.

##### 4.4.1.2. Without ICS Control - 177-FA NSS

Without ICS action, the neutron power does not run back after a reactor trip. The flux is reduced slightly due to moderator feedback, but the flux-to-RC

flow trip setpoint is reached in about 15 seconds, resulting in a reactor trip. POWER TRAIN results for this event are also shown in Figures 4-6 and 4-7.

#### 4.4.1.3. With ICS Control - 205-FA NSS

Figure 4-8 shows a 205-FA NSS one-pump coastdown with ICS control. A case without ICS would show the same relative differences as the 177-FA NSS comparison.

#### 4.4.2. Turbine Trip From 100% Power - 177-FA NSS

A turbine trip initiates a 20%/minute ramp decrease in ULD (see A<sub>2</sub> on Figure 4-3). This causes the core power and feedwater flow to decrease at about 20% per minute. However, because of variations in the core average coolant temperature ( $T_{avg}$ ) during the transient, the core power ramp rate varies somewhat from the 20% per minute rate. This happens because the variation in  $T_{avg}$  from its setpoint modifies the ULD signal to the control rods. The feedwater ramp rate also experiences some changes during the runback, which are caused by steam pressure variations. These variations occur when steam system valves open and close.

The excess steam produced during the runback is relieved through the main steam safety valves (MSSVs) until the production rate is less than the maximum atmospheric dump and condenser dump capacities. At this point, the MSSVs close, and all further steam production is controlled by the turbine bypass system (condenser dump and atmospheric vents). When the steam production decreases to the capacity of the condenser dump, the steam pressure steadies out at the modulating condenser dump setpoint with all steam bypassed to the condenser.

The runback terminates in about 5 minutes at about 15% power. Pertinent results for this transient are shown in Figure 4-9.

Turbine Trip - 205-FA NSS - The 205-FA NSS responds much like the 177-FA NSS during a turbine trip runback. The only major difference is the steam bypass route. The atmospheric vents and condenser dumps can accommodate all the excess steam production during the runback. Figure 4-10 shows the results for this transient.

#### 4.4.3. Reactor Trip From 100% Power — 177-FA NSS

Upon initiation of a reactor trip signal, the control rods drop into the core and core power decreases exponentially as shown in Figure 4-11. The ULD-to-feedwater control starts to decrease feed flow at 20% per minute; however, the Btu limits and cross limits on feedwater control quickly ramp main feedwater to a low value consistent with maintaining a minimum level in the steam generators. The core hot leg temperature decrease is the primary factor that decreases the Btu limits and reduces the feed flow rapidly.

Feedwater flow continues to decrease due to Btu limits until the steam generator liquid level reaches the low level setpoint. Beyond this time, the feedwater flow varies to maintain the steam generator at a minimum level (see  $\Delta_{12}$  in Figure 4-3).

The turbine trips when the reactor trips. The steam production in the first 5 to 10 seconds of the transient increases steam pressure and may cause the MSSVs to open. As the power (and consequently the steam production) decreases rapidly, the steam bypasses through the modulating condenser dumps. This allows steam pressure to steady out at the condenser dump setpoint. Figure 4-11 shows results for this transient.

Reactor Trip — 205-FA NSS — The response of the 205-FA NSS during a reactor trip is similar to that of the 177-FA NSS. The only major difference is the steam bypass route. The atmospheric vents and condenser dump valves can accommodate all the excess steam production during the trip transient, and the MSSVs will not open. Figure 4-12 shows results for a reactor trip transient (including a single stuck control rod).

#### 4.5. Sensitivity Studies

Initial condition variations have been selected to show the sensitivity of 177- and 205-FA NSS POWER TRAIN transient results to these variations. The following parameters and variations have been selected:

<u>Parameter</u>	<u>Variation from nominal</u>
RC flow	-5%
Pressurizer surge line $\Delta P$	+25%
Pressure level	-50 inches
RC pressure	-50 psi

<u>Parameter</u>	<u>Variation from nominal</u>
Core T <sub>avg</sub>	+2F
Turbine header pressure	+25 psi

These parametric variations were applied to a one-RC pump trip or reactor trip transient on the 177- and 205-FA NSS POWER TRAIN models.

#### 4.5.1. POWER TRAIN Sensitivity to Primary System Inputs

##### 4.5.1.1. Reactor Coolant Flow

Should RC flow input to POWER TRAIN vary 5% below the design value (or plant measured data), the enthalpy rise across the core will increase 5%, the cold leg temperatures will decrease slightly, and the hot leg temperature will increase slightly. This variation will have an insignificant effect on all POWER TRAIN parameter responses during a typical transient.

##### 4.5.1.2. Surge Line $\Delta P$

The effect of a 25% increase in surge line  $\Delta P$  on a typical transient is insignificant. For example, during a reactor trip, the increase in surge line  $\Delta P$  causes a slight lag in pressurizer level decrease/increase. This in turn causes an RC pressure variation of about 10 psi (compared to nominal surge  $\Delta P$  during a trip). The responses of the other primary/secondary parameters do not vary.

##### 4.5.1.3. Pressurizer Level

An initial pressurizer level of 50 inches less than nominal influences two RC parameters. The pressurizer level will be about 50 inches low throughout any transient, and because there is less initial total energy in the pressurizer, a lower RC pressure occurs during an outsurge transient. An initial level 50 inches low produces a maximum RC pressure that is approximately 50 psi lower during a trip transient.

##### 4.5.1.4. RC Pressure

A 50 psi variation in initial RC pressure does not change any other steady state parameters significantly. During a reactor trip, the RC pressure varies continuously through the transient at about 50 psi low compared to the nominal case. The other primary/secondary parameters will not vary significantly.

#### 4.5.1.5. Core $T_{avg}$

A 2°F increase in the core average temperature increases the cold and hot leg temperatures by 2°F. During a one-RC-pump coastdown transient, the initial  $T_{avg}$  increase causes the RC pressure to reach the pressurizer PORV setpoint a few seconds sooner (compared to the nominal  $T_{avg}$  transient). The minimum RC pressure during the transient is approximately the same in both cases. No other primary system parameters change.

#### 4.5.1.6. Turbine Header Pressure

A 25 psi initial increase in the turbine header pressure causes the RC average fluid temperature to increase slightly during a one-RC-pump coastdown. However, since the power runback is faster (due to the moderator feedback effect), the overall effect of RC pressure is reduced. The 205-FA NSS will still open the pressurizer PORV, and the 177-FA NSS peak pressure will be about 20 psi lower.

Table 4-1 shows the results of these variations compared to a nominal design transient POWER TRAIN case. The results show that the magnitude and direction of the pertinent transient responses are predictable and can be used to establish limiting conditions from parametric studies.

### 4.6. POWER TRAIN Vs Plant Data Vs Test Data

#### 4.6.1. Test Data Comparison

In 1978, B&W conducted a series of tests at the Alliance Research Center (ARC) to obtain additional performance data on the OTSG for computer model development and verification. The tests, selected to investigate loss-of-feedwater events, were performed on a 19-tube model steam generator operated in the following configurations:

205-FA NSS — IEOTSG — 3600 MWt configuration

205-FA NSS — IEOTSG — 3800 MWt configuration

— 600F  $T_{avg}$ , 1050 psia steam

— 595F  $T_{avg}$ , 1050 psia steam

— 595F  $T_{avg}$ , 1000 psia steam

The steam generator model described in sections 2.16 and 2.17 has been compared with the test data as shown in Figures 4-13 through 4-16. These data show excellent agreement between the test data and the simulation.

#### 4.6.2. POWER TRAIN Benchmark Transients

This section describes comparisons of POWER TRAIN responses to data from 177-FA operating plants. The validity of the modeling is established by evaluating the similarities and differences between measured and calculated quantities. Two transients are presented — a turbine trip with runback and a reactor coolant pump trip. Each transient is discussed in two parts.

##### 4.6.2.1. Turbine Trip Transient

The turbine trip transient with runback was initiated on an operating 177-FA NSS plant as a test of the ability of the unit to respond automatically to an unexpected loss of load. The objective was to ramp power from 100 to 15% full power at 20% per minute without reaching reactor protection system (RPS) set-points. The transient response was recorded by the B&W Reactimeter (data logging device) and Brush recorders.

Figure 4-18 compares (plant data Vs POWER TRAIN) several variables for the turbine trip transient including fluid temperatures, pressure, and flow. Since the transient is symmetric, conditions are presented for only one loop.

In broad perspective, the comparison is quite good. The general trends and sequence of events are reproduced quite well. Moreover, the comparison is not "retouched." Neither the model nor the data were modified or adjusted to improve the agreement.

The following paragraphs discuss differences between the plant and the model. The discussion is organized by grouping together differences that are linked to a common cause.

Initial conditions — Ideally, the plant and model start a benchmark transient from exactly the same initial state. In this benchmark, the two differed slightly. The most significant difference is in the reactor coolant inlet and outlet temperatures. The plant was not at perfect steady-state conditions at the beginning of the transient, while the POWER TRAIN model more closely approximated perfect steady-state conditions at event initiation.

Steam Line Noding — The steam line model lumps the main steam line from the safety relief valves to the turbine header into a single parameter — steam volume. The lumped modeling follows the average behavior of the steam volume. The plant safety and relief valves open following turbine trip when the first

pressure wave hits them. The relief valves open slightly later when the average steam pressure reaches their setpoints. Earlier opening of the plant relief valves causes the pressure rise at the generator to be less severe, and the corresponding rise in RC inlet temperature is slower.

The model operates at a higher average pressure in the time period between 10 and 70 seconds despite having a lower reactor power and lower feedwater flow rate. This is probably because the plant has more installed bypass capacity than the POWER TRAIN model (25 Vs 15%).

The intermittent operation of one bank of POWER TRAIN valves in the time interval from 10 to 70 seconds is a result of the relatively small (25 psi) reset bias. This intermittent operation causes a sawtooth-like pressure response. This effect propagates to the rest of the model. It is particularly evident for feedwater flow and reactor coolant inlet temperature. The plant relief valves do not open and close repeatedly; hence, pressure response is smoother.

The higher average secondary pressure response of the model gives a higher temperature on the secondary side of the steam generator than in the plant. Consequently, reactor coolant inlet temperature is higher on the model primary side. This higher inlet temperature contributes directly to lower model power through negative power feedback of the moderator temperature coefficient. The higher moderator temperature error also gives a larger  $T_{avg}$  correction to the control rod demand.

Feedwater Control Differences — The ICS at the plant has an additional feed pump control circuit (not modeled in POWER TRAIN) that increases pump speed following the turbine trip. The effect of the circuit is evident from the higher feedwater flow observed from 50 to 120 seconds. The higher flow rate contributes to the lower reactor coolant inlet temperature and higher reactor power observed in the plant data. The additional flow is not evident in the steam pressure since the additional relief capacity allows for higher flow at lower steam pressure.

Summary of Comparison — The model and the plant data show good agreement in magnitude, direction, and rate of change of the parameter responses. Since this agreement is the major criterion for the "unretouched" model, the model criteria are met.

#### 4.6.2.2. Reactor Coolant Pump Trip

An unplanned RC pump trip occurred at a 177-FA plant operating at approximately 90% power. Data from this event were obtained by the B&W Reactimeter (data logger). The control and performance objective of the NSS following an RC pump trip is to reduce power at rates up to 50% per minute from the initial power to the operating power level limit of 75% full power without a reactor trip.

The plant data and POWER TRAIN data are compared for this event. The POWER TRAIN model is not "retouched," i.e., no modifications were made to adjust the POWER TRAIN controls to the exact settings of the specific operating plant.

The POWER TRAIN response is compared to the plant data in Figure 4-19. In general, the POWER TRAIN response comparison to the plant data is good. The overall trends during the transient are similar, and the new steady-state operating condition comparisons are very good. Specific details of the transient are described below.

Reactivity Worth of Rods — The dominant response difference (plant Vs model) on the primary side is that reactor power is reduced faster in the plant. Since the control rod speed is the same in the model and the plant, the most likely cause of the difference is the worth of the control rods. The lower power produces a slower rise in primary temperature at the core outlet. The reduction in reactor power at the plant provides matching of heat removal from primary to secondary within 18 seconds. From 18 to 44 seconds, heat removal exceeds generation, and primary temperatures and pressure decrease. At 44 seconds, the average reactor coolant temperature drops below 579F. Since the average reactor coolant temperature is a strong term in flux error control, rods are withdrawn to increase power and to return the average temperature to setpoint values. During the same 44-second period in the model, the reactor power remains higher than the secondary heat removal. Consequently, the model's average temperature and pressure rise throughout the period.

Feedwater Flow — The time-dependent behavior of feedwater flow is somewhat different between the plant and the model, particularly for loop B. The loop B feedwater flow in the plant increases slowly for 12 seconds, then begins to rise sharply. The loop B feed flow in the model begins to rise sharply at 4 seconds. Information from the plant indicates that the feedwater control valve

in loop B was initially sluggish in response. This accounts for the major differences (plant Vs model) in the loop B feedwater flow.

Summary of Comparison — The important plant-to-model comparisons are good. The plant and model respond in the same direction initially, and the control systems take the same corrective action automatically. Certain parameters, such as rod worth and component (valve) responses, are different from those in the not "retouched" model, and as a result, differences are to be expected.

Table 4-1. Sensitivity Study Results

Transient	Maximum/minimum parameters for nominal case			Initial condition parameter change	Maximum/minimum values of parameters during transient			Maximum change from base case		
	Parameter	177-NSS	205-NSS		Parameter	177-NSS	205-NSS	177-NSS	205-NSS	
	RC pressure, psia	2170/1730	2250/1970	Pressurizer surge line $\Delta P$ increased 25%	RC pressure	No change	2250/1960	0/0 psi	0/-10 psi	
				Pressurizer level	No change	210/110 in.	0/0 in.	0/-10 in.		
				Initial pressurizer level reduced 50 in.	RC pressure	2170/1680	2250/1940	0/-50 psi	0/-30 psi	
				Pressurizer level	220/40	160/60	-50/-40 psi	-50/-60 psi		
Reactor trip from 100% power	Pressurizer level, in.	270/80	210/120	Initial core pressure reduced 50 psi	RC pressure	2120/1700	2200/1940	-50/-30 psi	-50/-30 psi	
One RC pump coastdown from 90% power	RC pressure, psia	2270/2430	2360/2070	Initial core $T_{avg}$ +2F high	Pressurizer level	No change	No change	0/0 in.	0/0 in.	
					RC pressure	No change	No change	0/0 psi	0/0 psi	
					RC core $T_{avg}$	589/584	605/597	+3/+2F	+3/+2F	
					RC pressure	2260/2190	2360/2100	-10/+10 psi	0/+30 psi	
	RC $T_{avg}$ , F	586/582	605/592	Initial and coastdown RC flow rates 5% less	RC core $T_{avg}$	587/582	604/591	+1/0F	-1/-1F	
					RC pressure	2230/2130	No change	-40/0 psi	0/0 psi	
					Initial turbine header pressure 25 psi high	RC core $T_{avg}$	No change	No change	0/0F	0/0F

Figure 4-1. Typical POWER TRAIN  
Noding Diagram

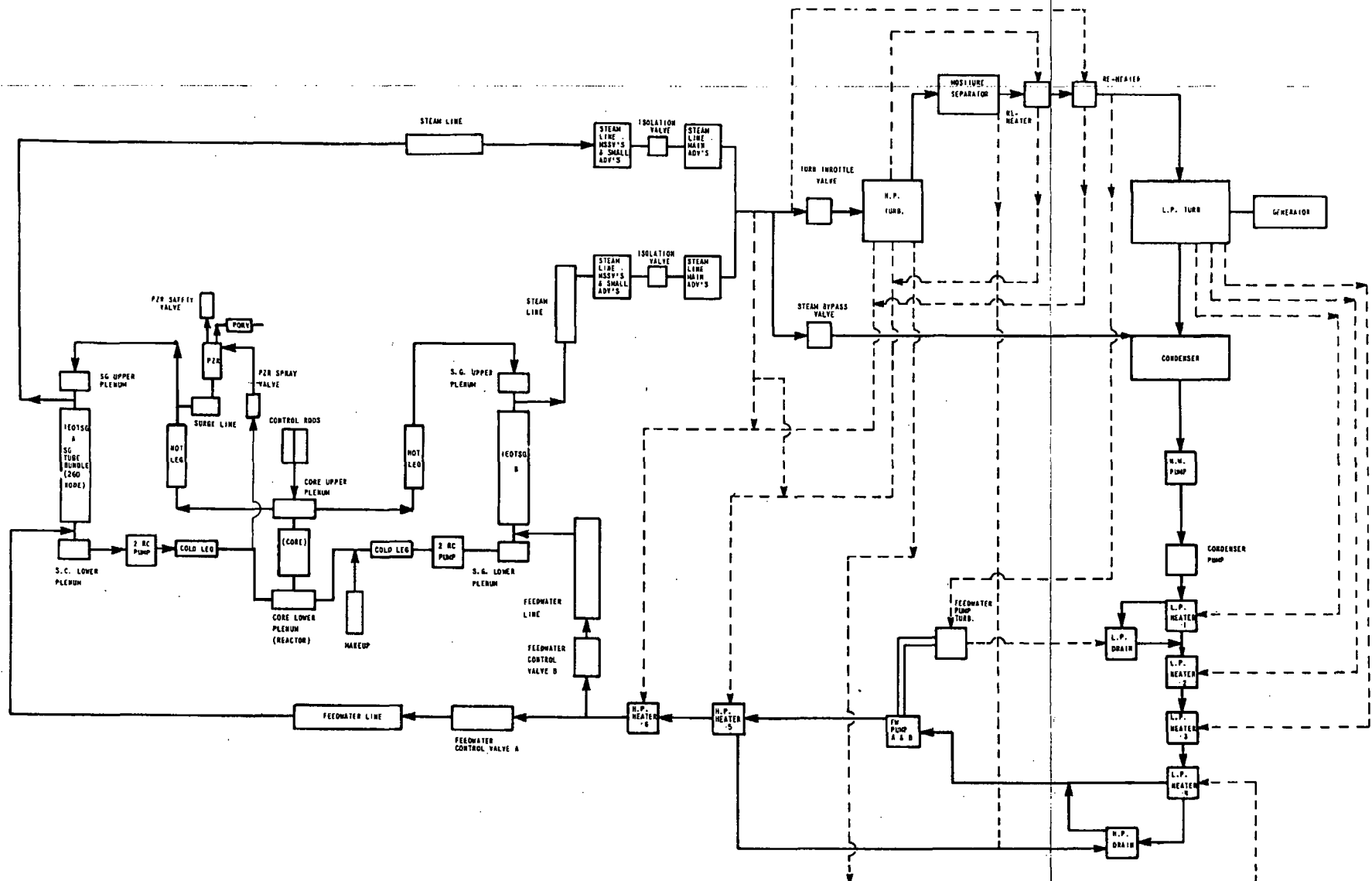
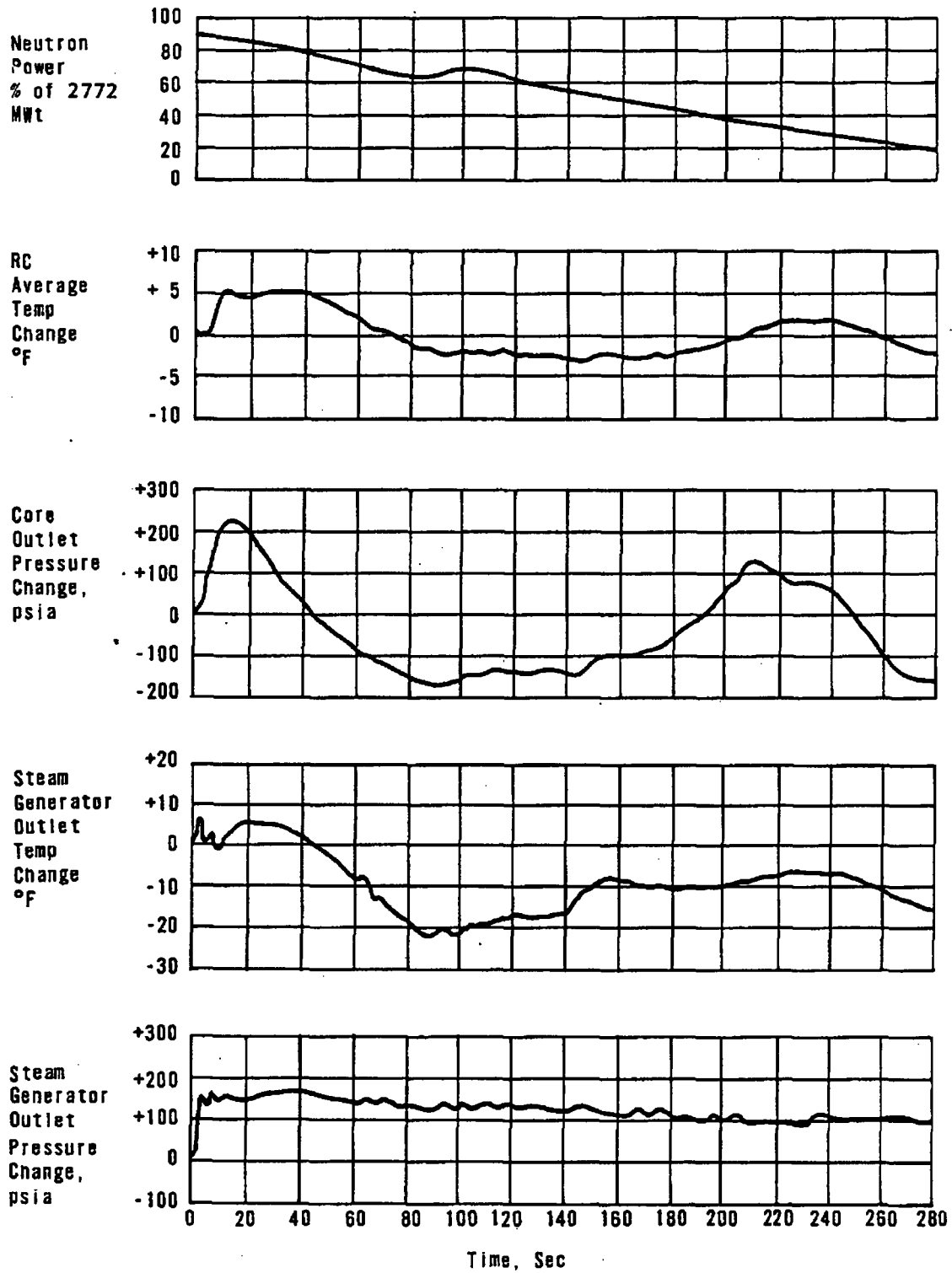


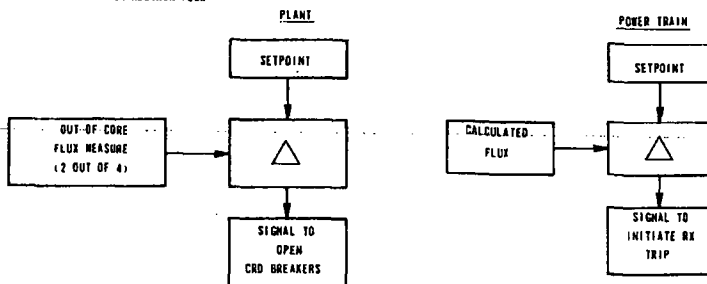
Figure 4-2. Turbine Trip With Runback - 177-FA NSS



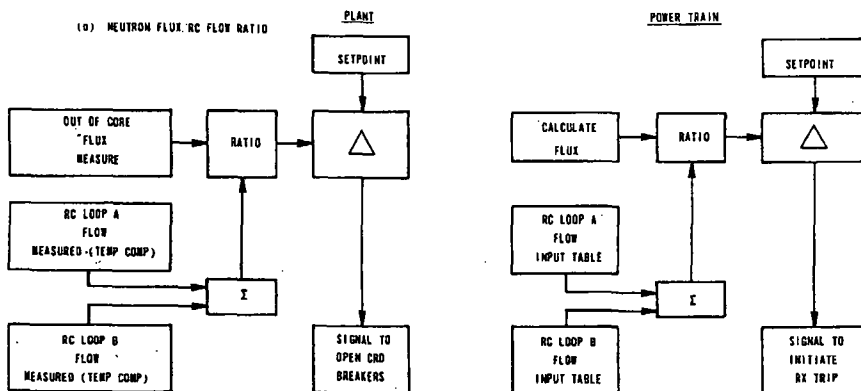
MODIFICATION  
 S = SWITCH  
 V = VOLTAGE GENERATOR  
 T = TUNER  
 TT = TURBO TRIP  
 A = DELTA  
 K/2 = PROPORTIONAL + INTEGRAL  
 PROCESSING OF SIGNAL  
 N/A = UNKNOWN STATUS  
 - = INVERTER  
 F(D) = FUNCTION OF DOWND  
 F(T) = FUNCTION OF TIME  
 > = GREATER THAN  
 < = LESS THAN  
 > < = 2 LINES  
 \* = MULTIPLY  
 ◇ = CONNECTING POINTS  
 + = CONNECTION  
 - = NO CONNECTION

F<sub>1</sub> = FREQUENCY ERROR  
 N<sub>1</sub> = NEGATIVE MOMENT  
 P<sub>1</sub> = TURBINE HEADS PRESSURE  
 T<sub>1</sub> = TURBINE TEMPERATURE  
 F<sub>2</sub> = LOOP A FEEDWATER FLOW  
 F<sub>3</sub> = LOOP B FEEDWATER FLOW  
 T<sub>2</sub> = TURBINE A/B LOOP  
 T<sub>3</sub> = TURBINE B/C LOOP  
 T<sub>4</sub> = TURBINE C/D LOOP  
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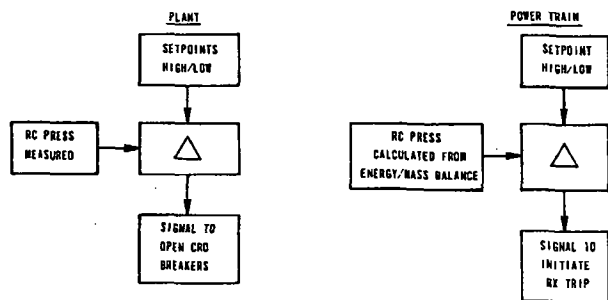
(b) HIGH NEUTRON FLUX



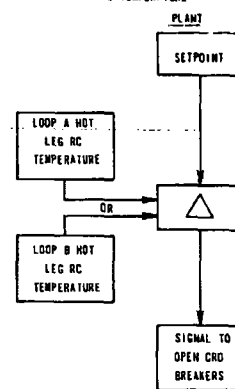
(c) NEUTRON FLUX/RC FLOW RATIO



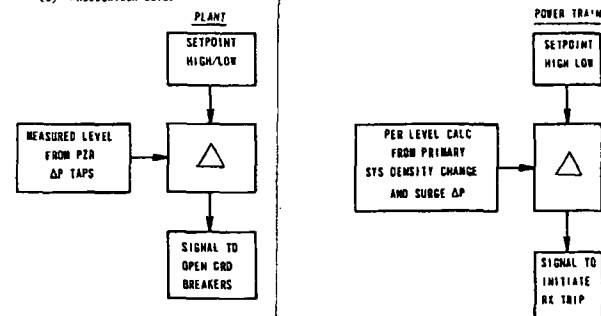
(d) HIGH LOW RC PRESSURE



(e) HIGH RC TEMPERATURE



(f) PRESSURIZER LEVEL



(g) CORE POWER DISTRIBUTION

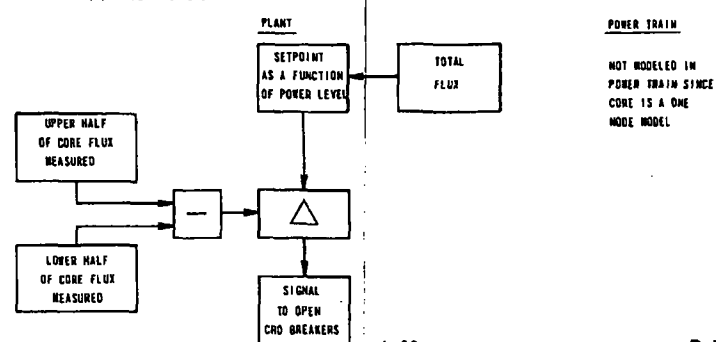
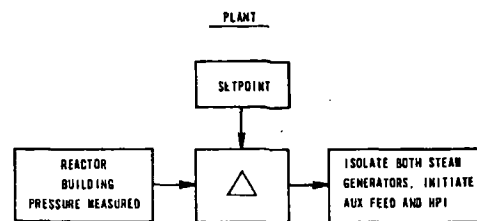
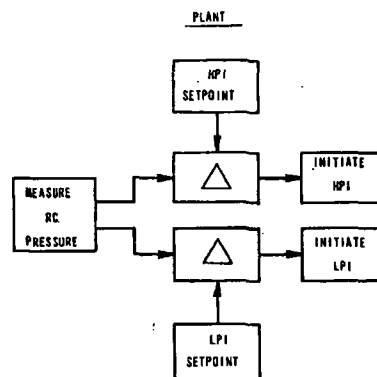
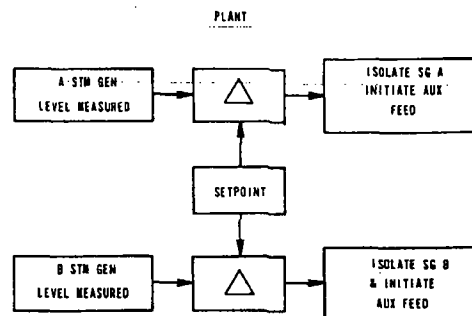
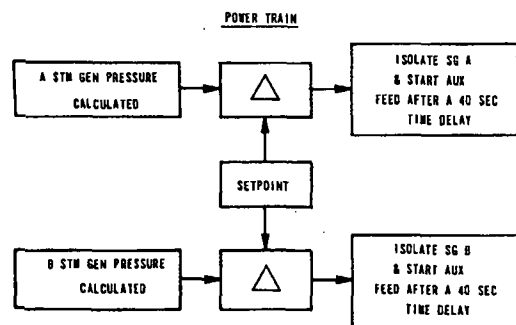
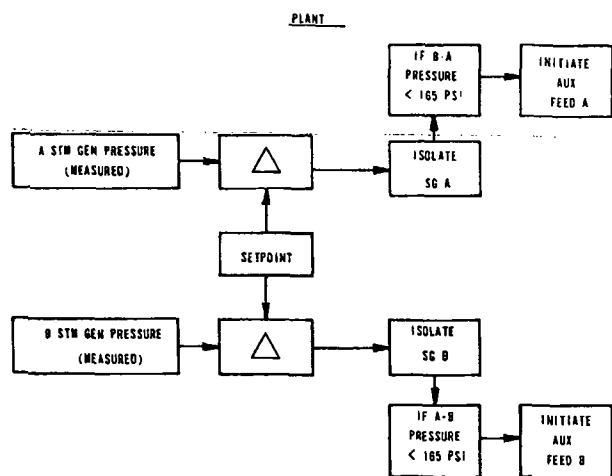


Figure 4-4. RPS Model in POWER TRAIN

Figure 4-5. ESFAS Model in POWER TRAIN



POWER TRAIN

NOT IN POWER TRAIN

POWER TRAIN

MAKEUP FLOW RATE CAN BE INCREASED TO SIMULATE HPI. NO LPI IS MODELED

POWER TRAIN

NO REACTOR BLDG MODELED IN POWER TRAIN

Figure 4-6. One-RC-Pump Coastdown From 90% Power With and Without ICS - 177-FA NSS

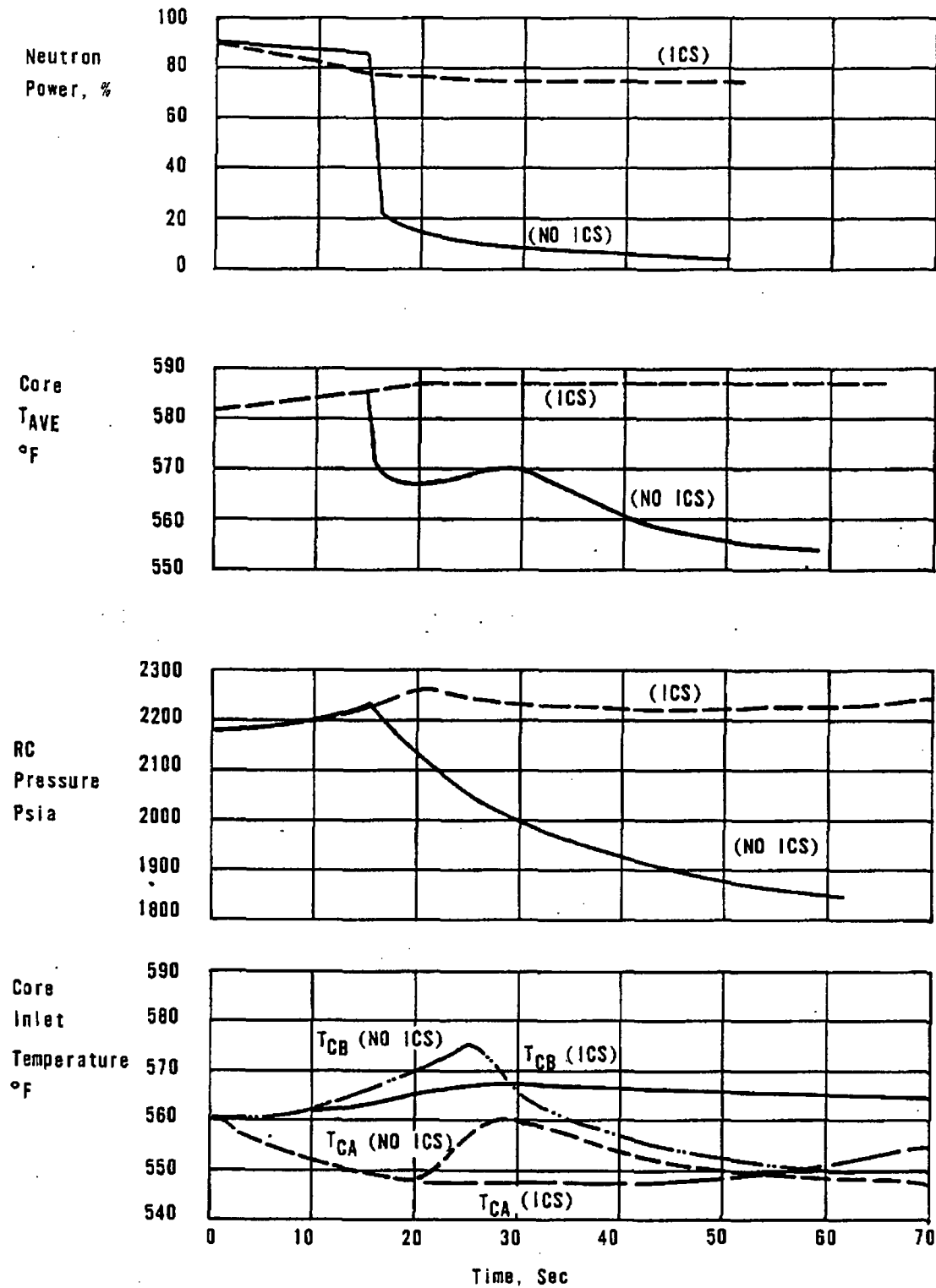


Figure 4-7. One-RC-Pump Coastdown From 90% Power  
With ICS - 177-FA NSS

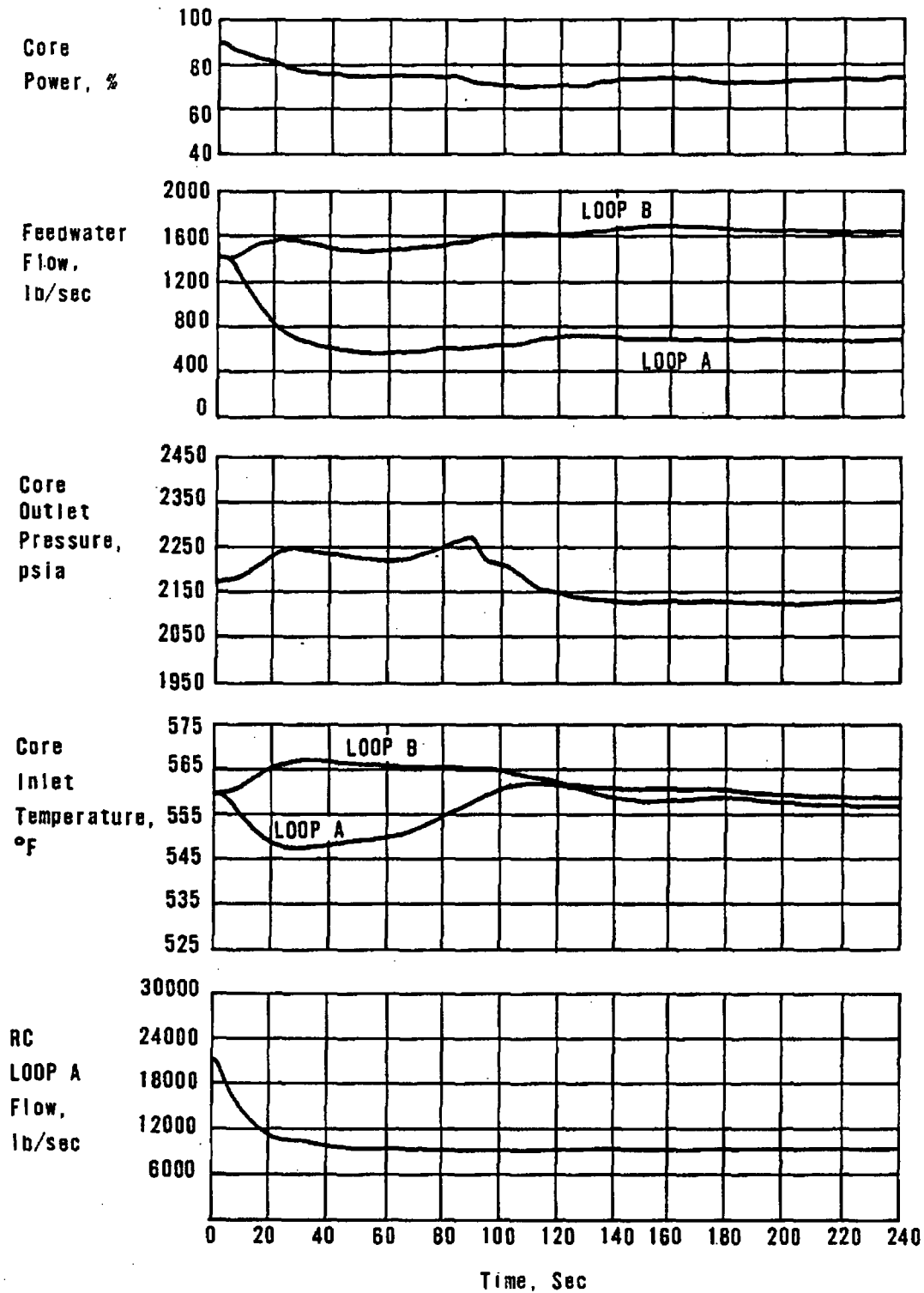


Figure 4-8. One-RC-Pump Coastdown From 90% Power — 205-FA

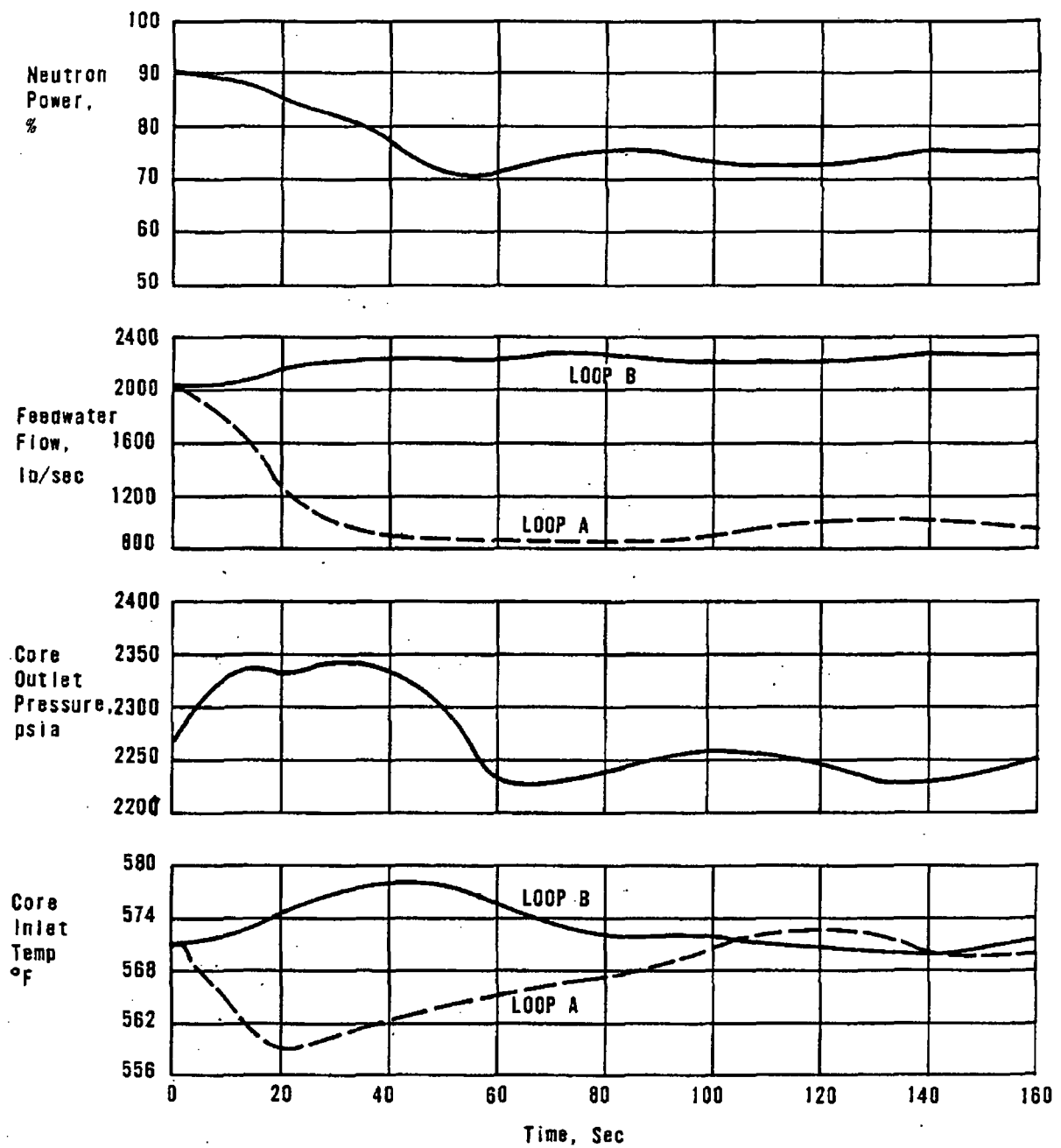


Figure 4-9. Turbine Trip From 100% Power —  
177-FA NSS

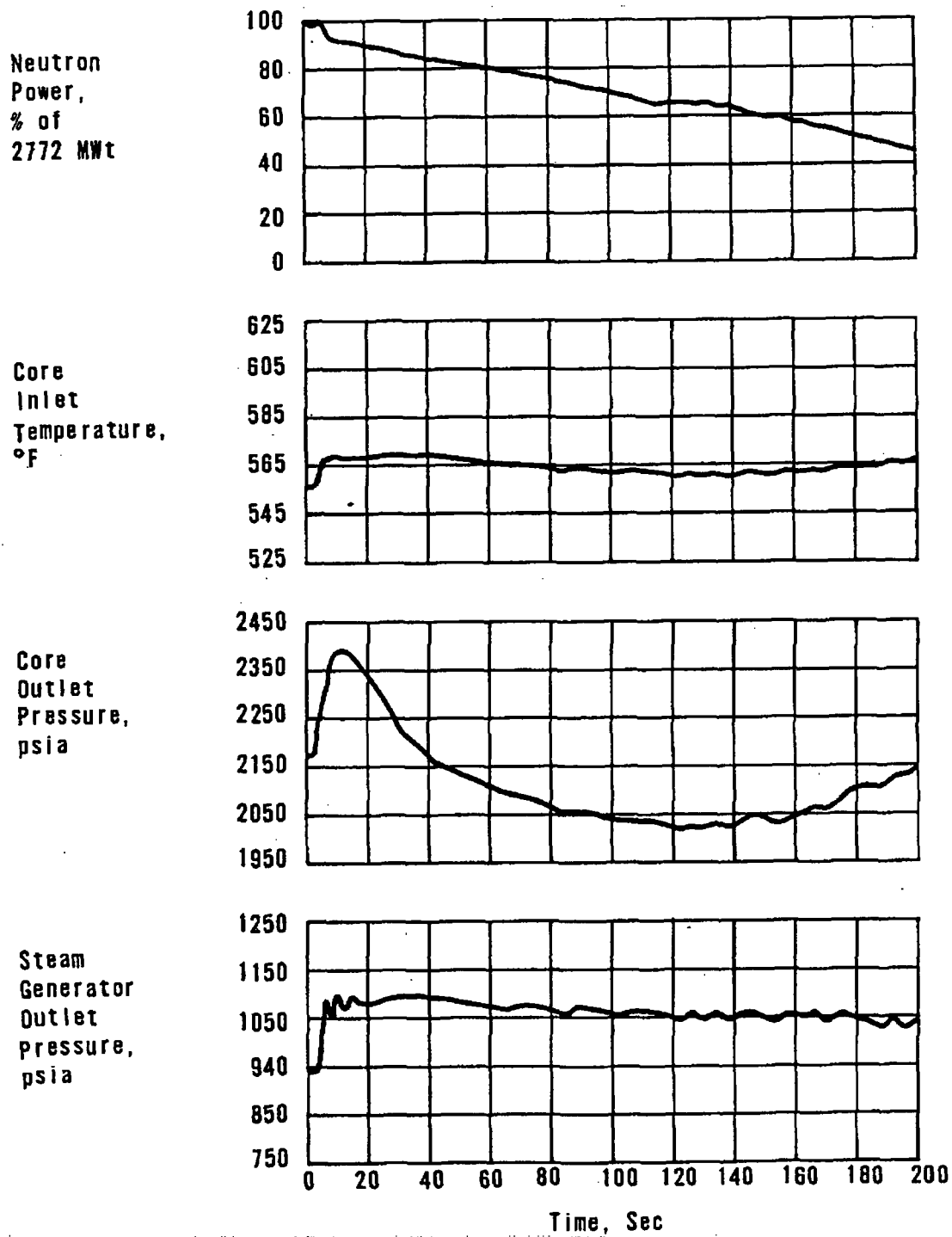


Figure 4-10. Turbine Trip From 100% Power — 205-FA NSS

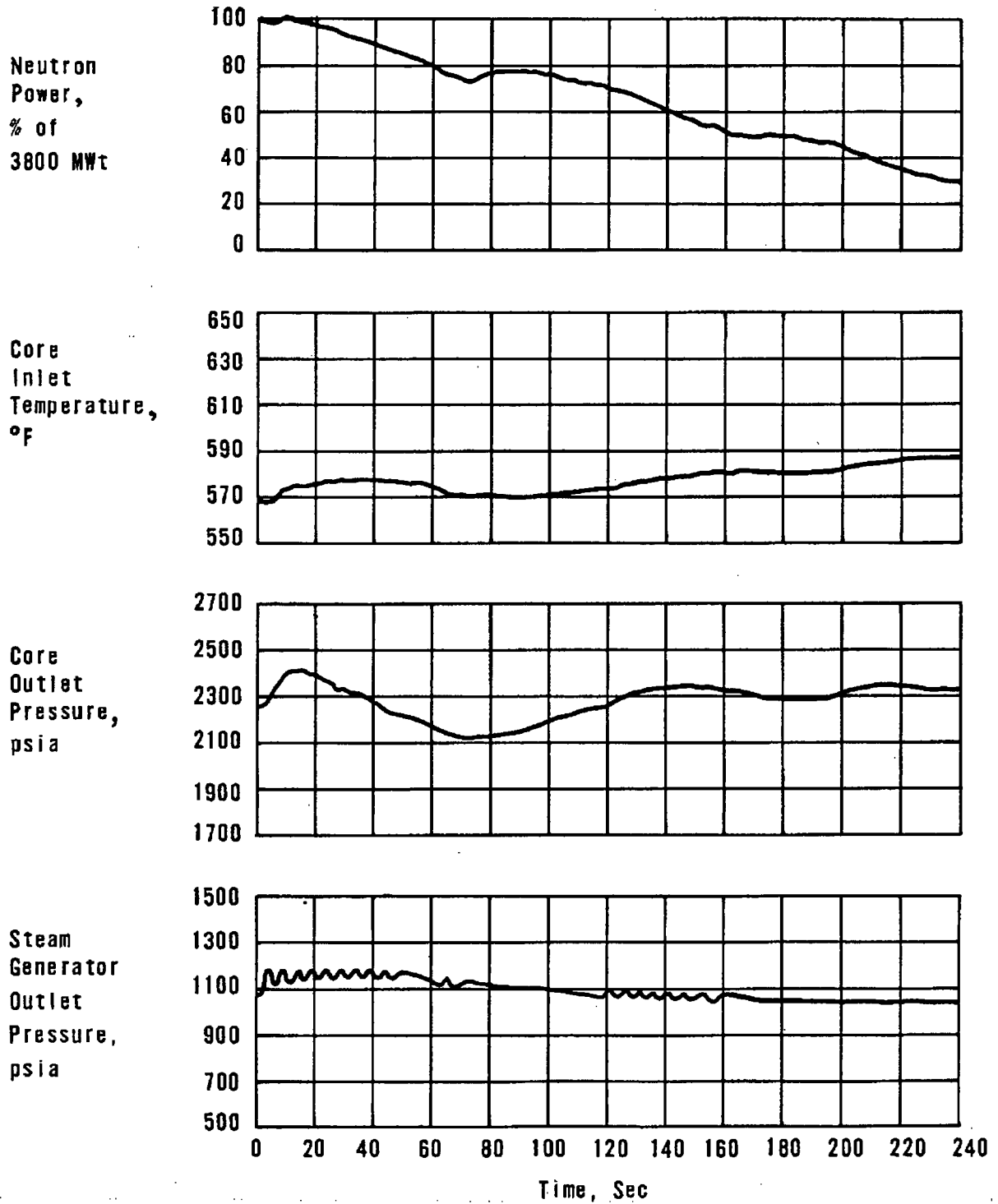


Figure 4-11. Reactor Trip From 100% Power - 177-FA NSS

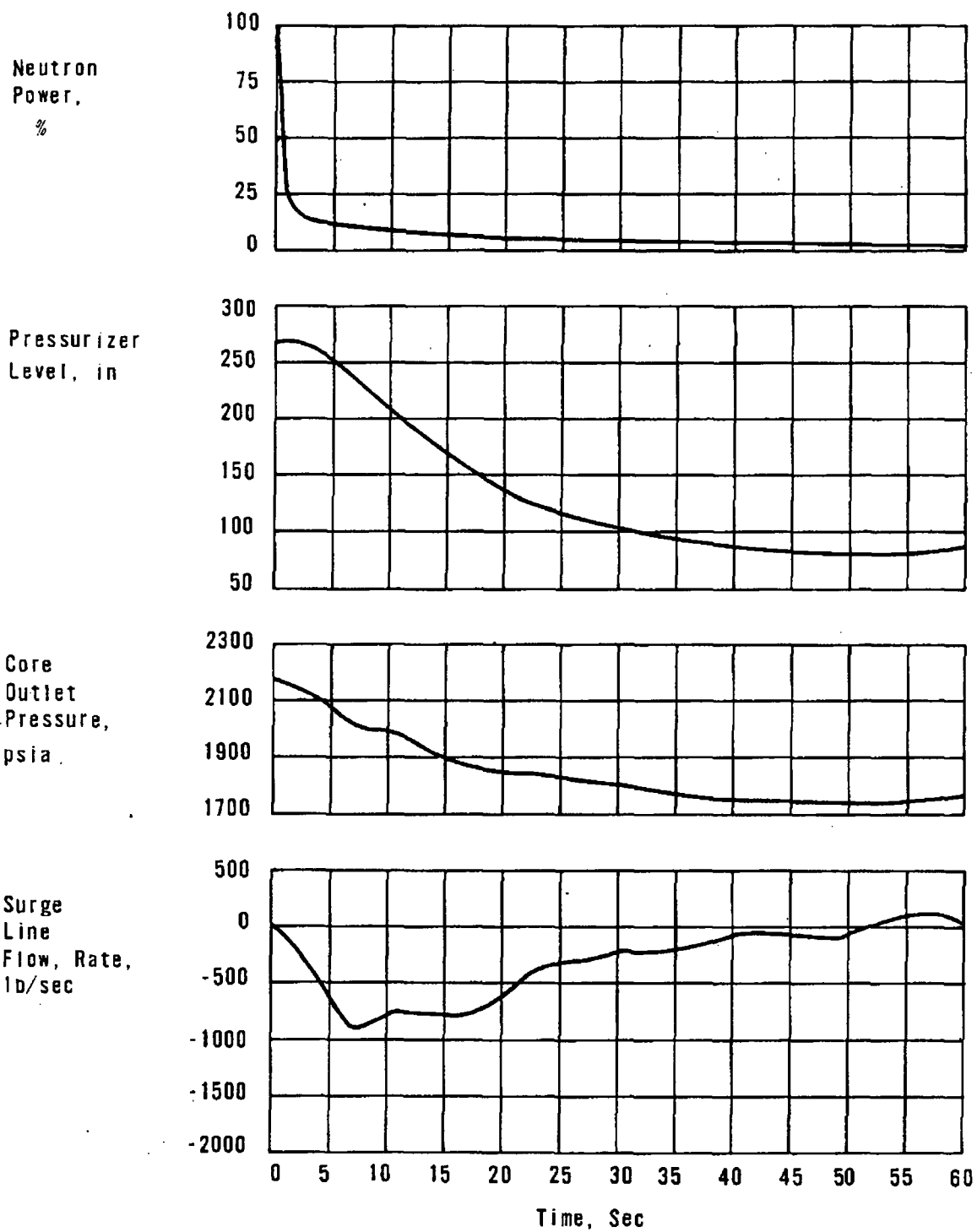


Figure 4-12. Reactor Trip From 100% Power - 205-FA NSS

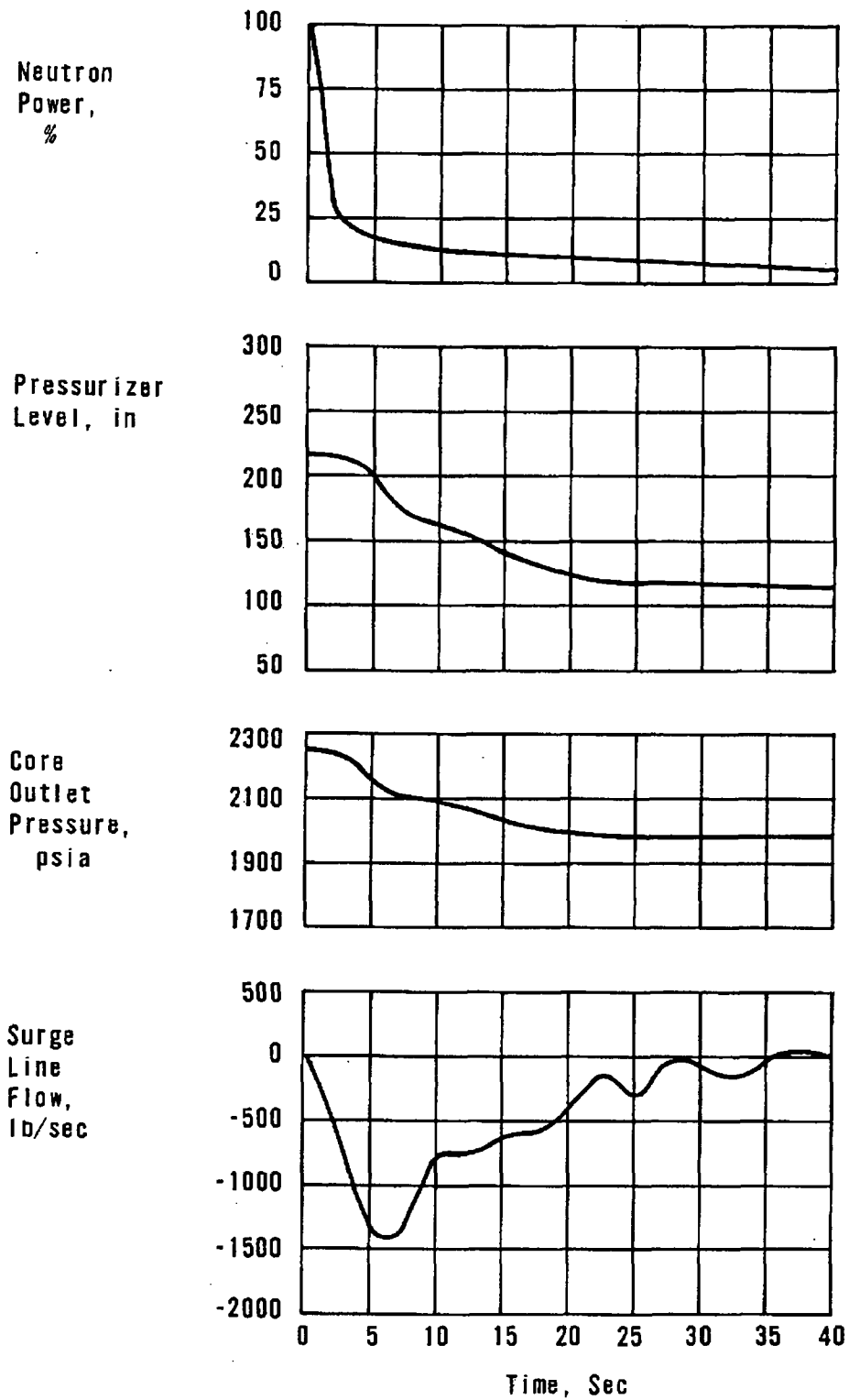


Figure 4-13. Feedwater Step to Produce 100-20% Steam Flow and Recovery (205-FA NSS, 1000 psi Steam, 3800 MWt)

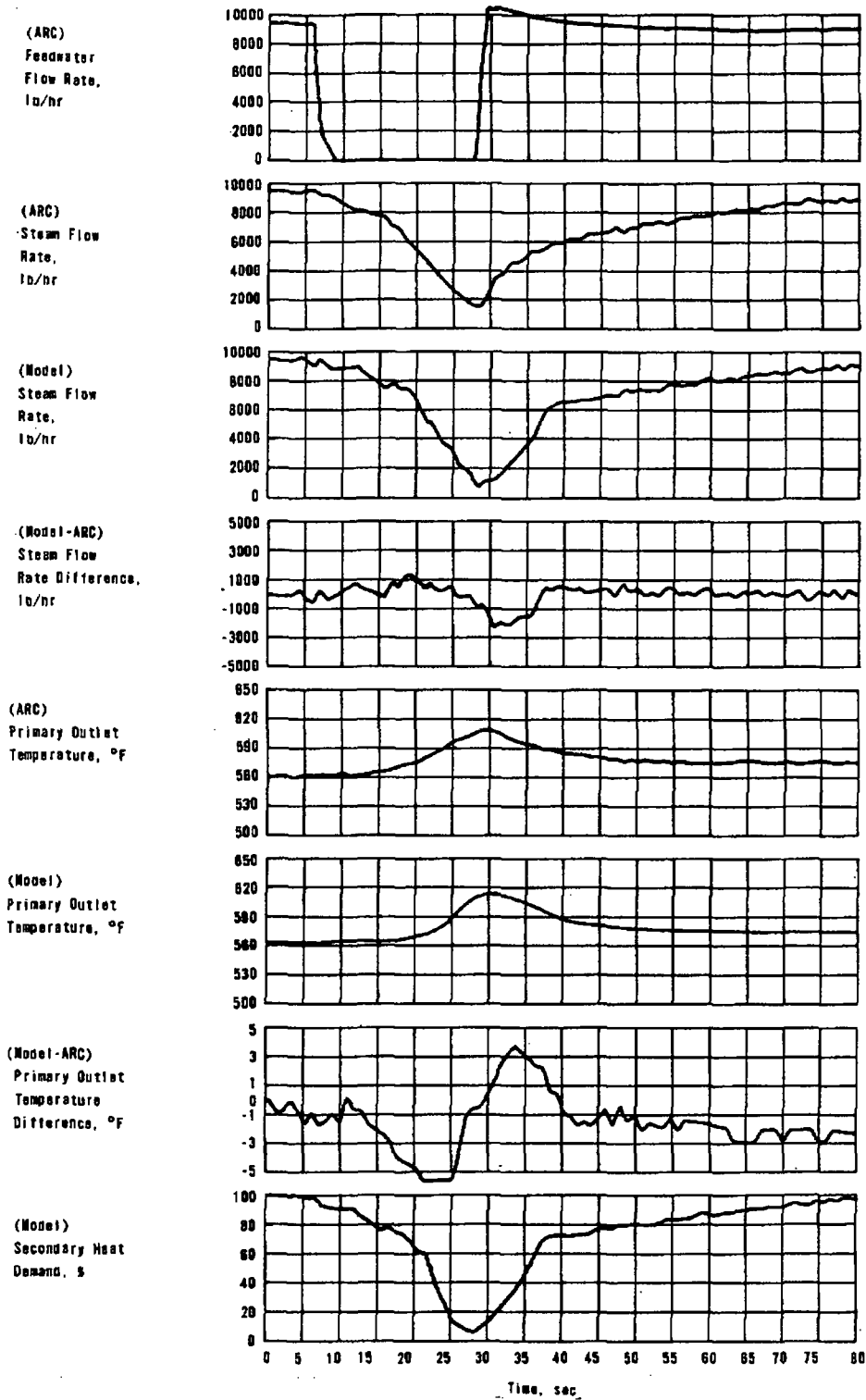


Figure 4-13. (Cont'd)

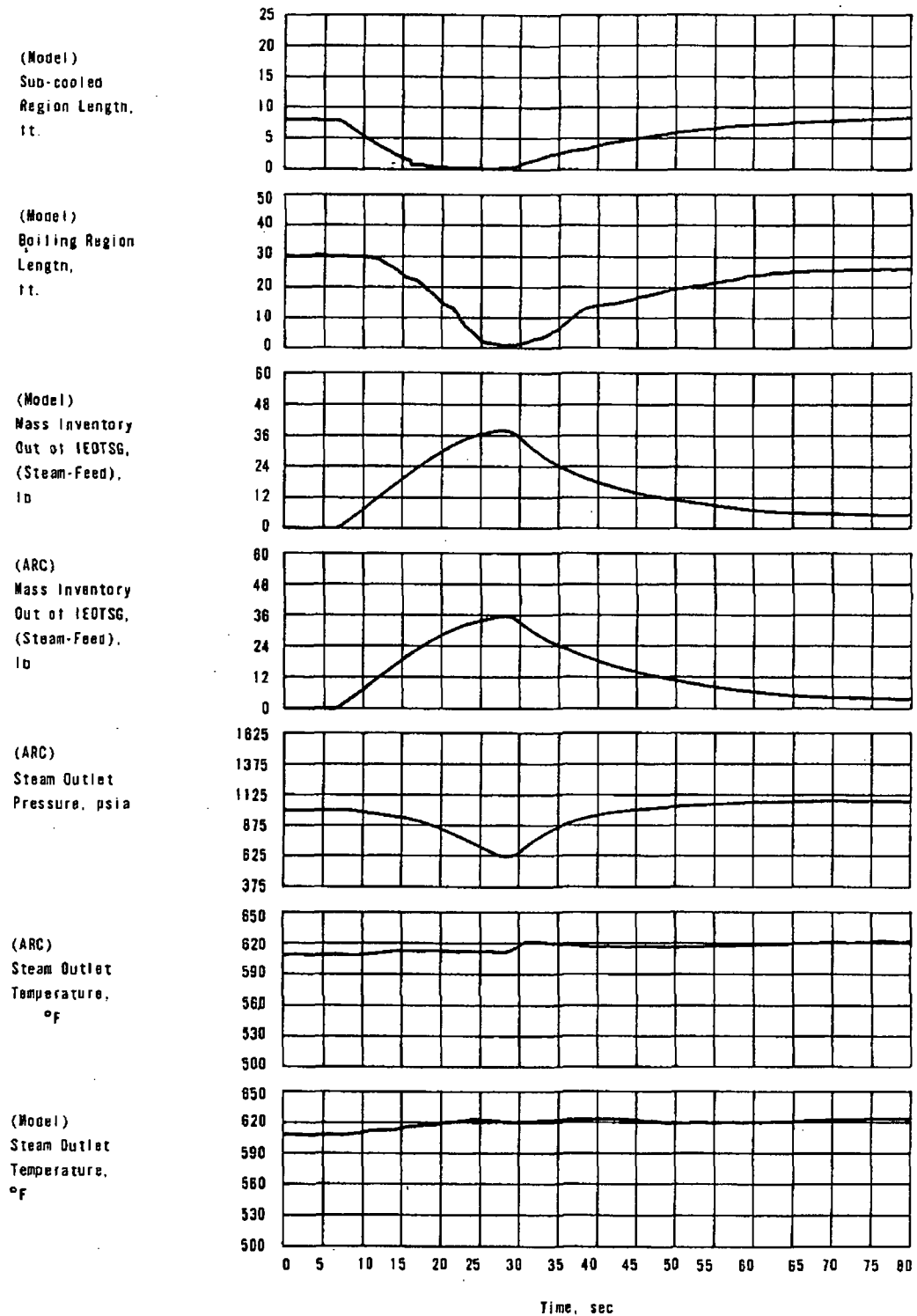


Figure 4-14. Feedwater Step to Produce 100-10% Steam Flow and Recovery  
(205-FA NSS, 1000 psi Steam, 3800 MWt)

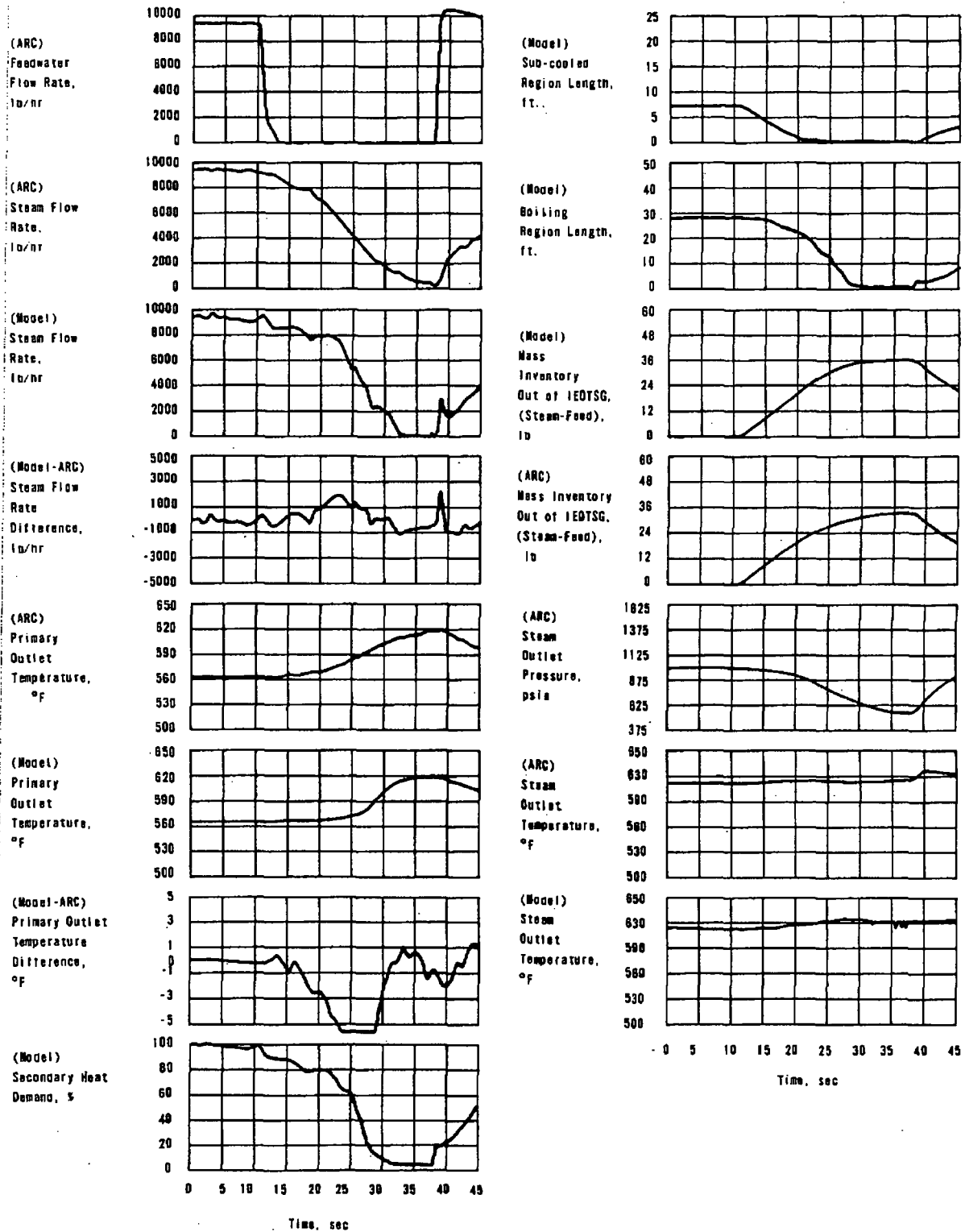


Figure 4-15. Feedwater Step to Produce 100-0% Steam Flow and Recovery (205-FA NSS, 1000 psi Steam, 3800 MWt)

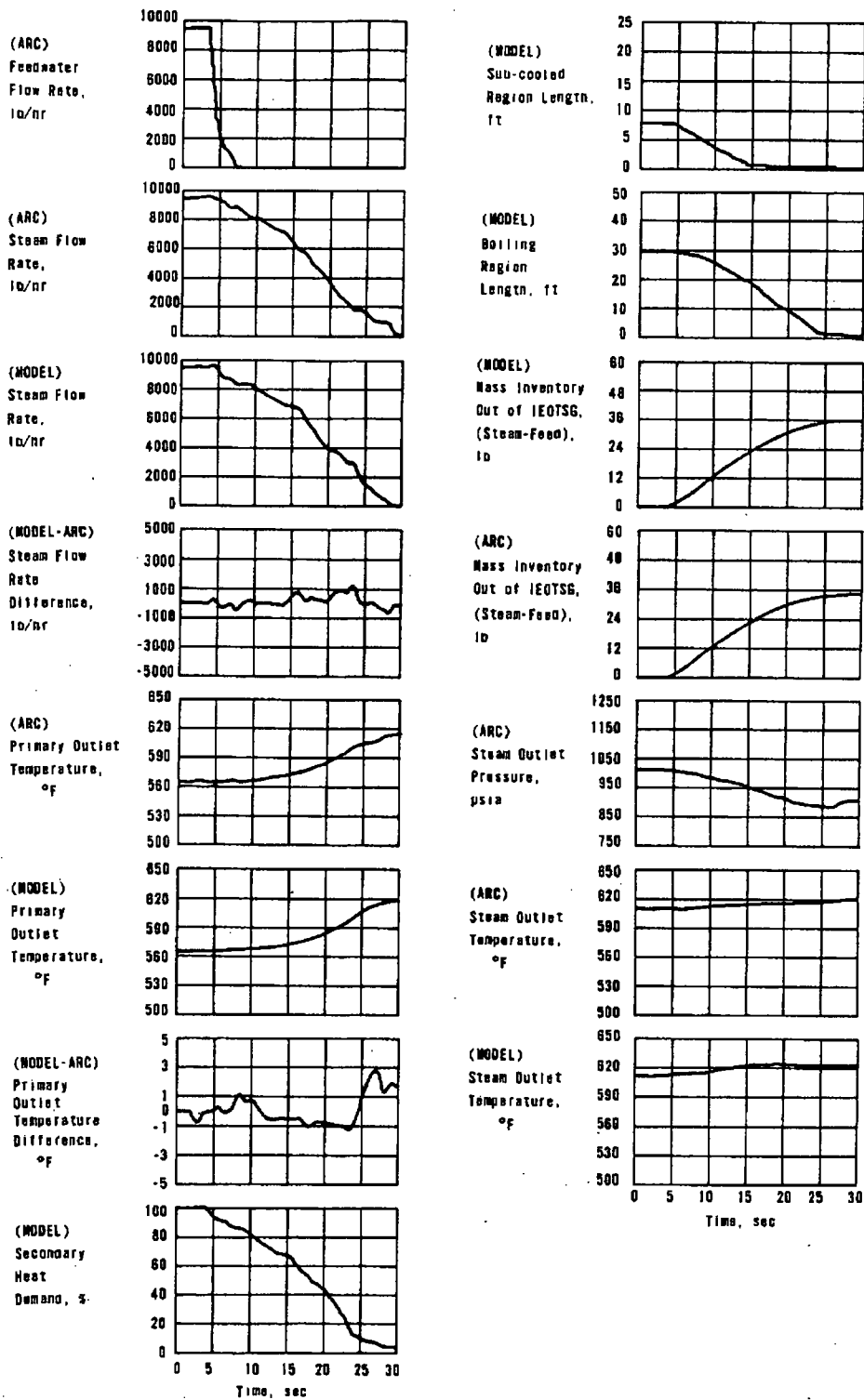


Figure 4-16. Feedwater Step to Produce 100-0% Steam Flow and Recovery (205-FA NSS, 1050 psi Steam, 3600 MWt)

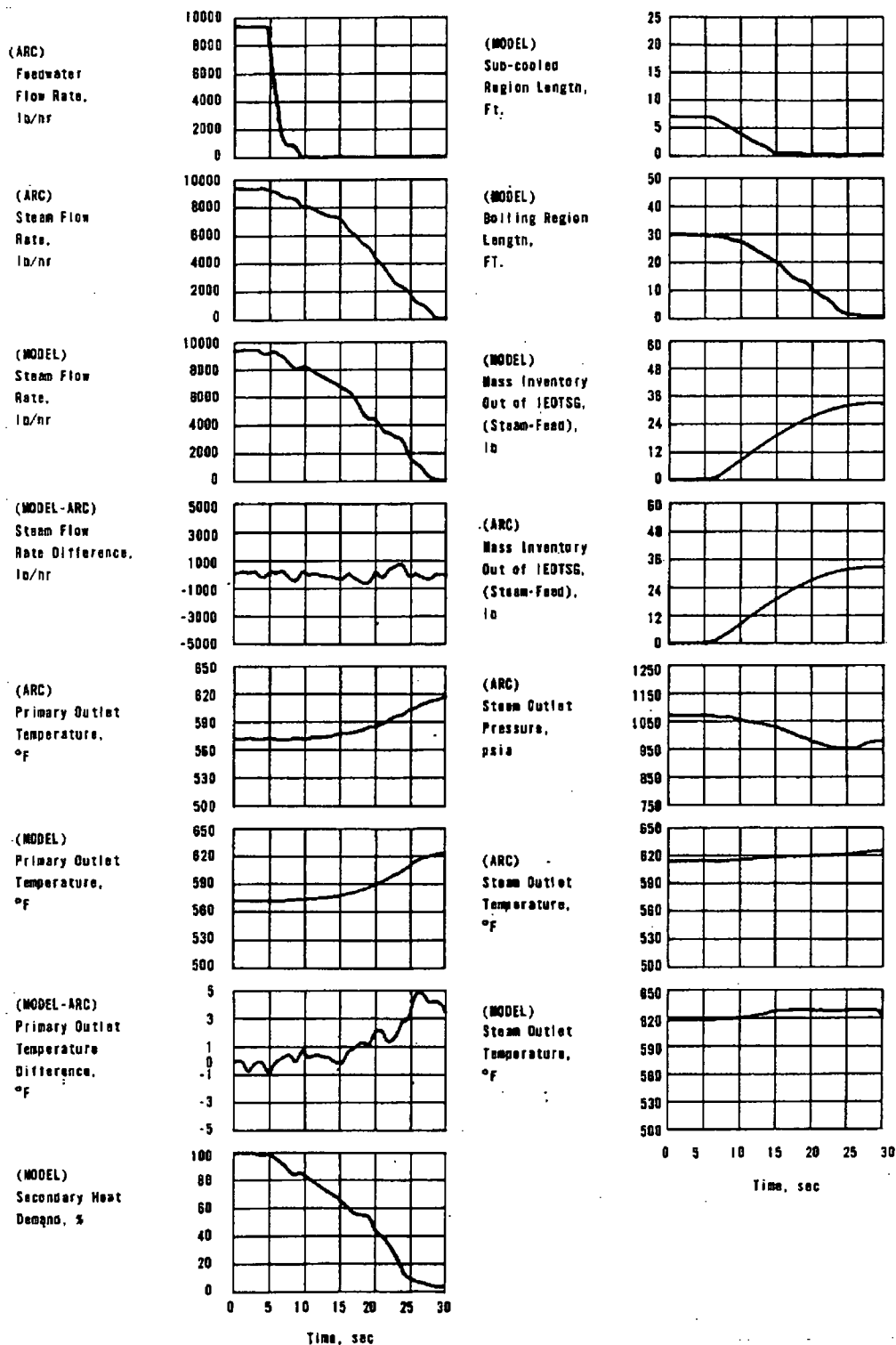


Figure 4-17. Feedwater Step to Produce 100-0% Steam Flow and Recovery (205-FA NSS, 1050 psi Steam, 3800 MWt)

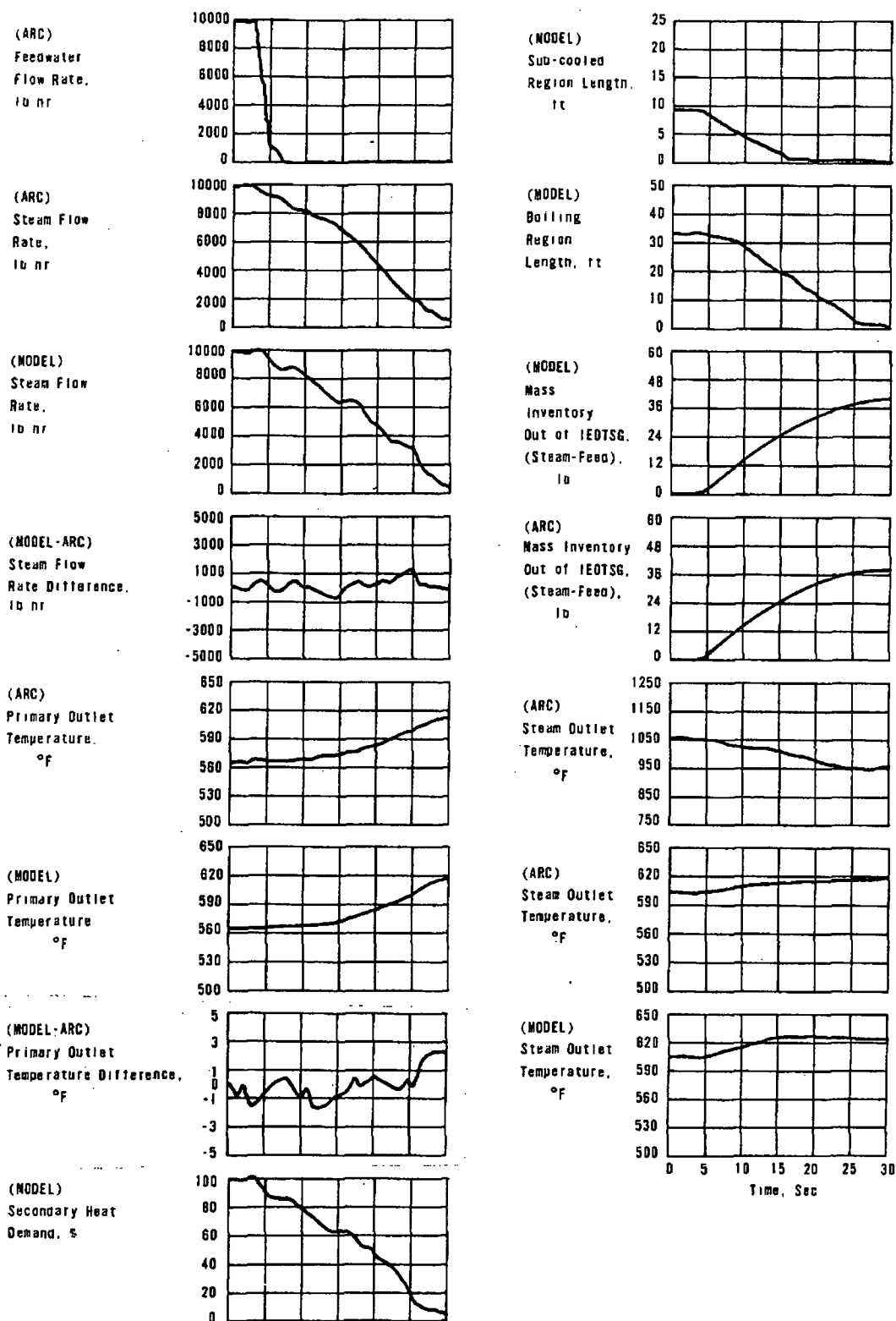


Figure 4-18. Plant Data Vs POWER TRAIN Data for  
Turbine Trip - 177-FA NSS

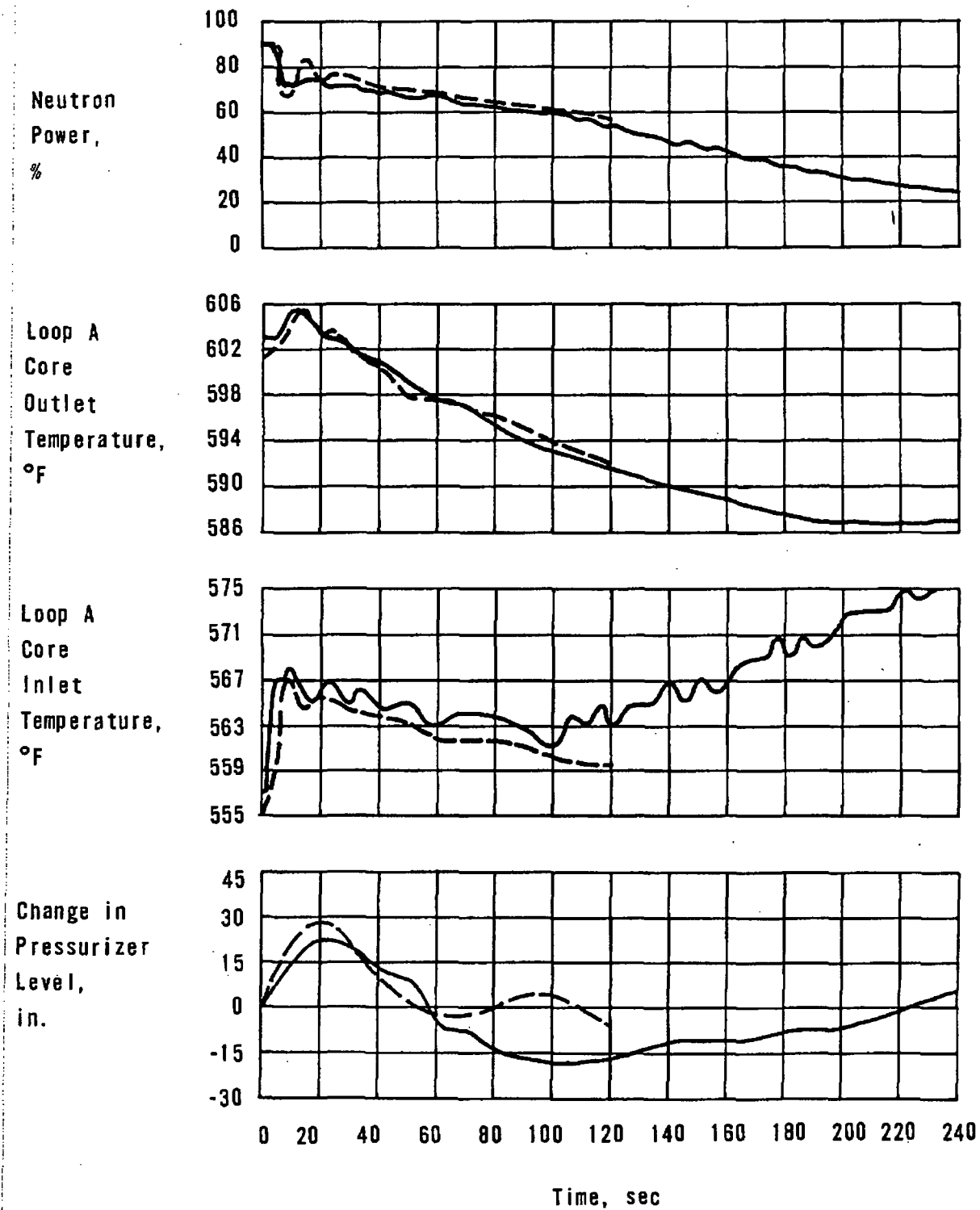


Figure 4-18. (Cont'd)

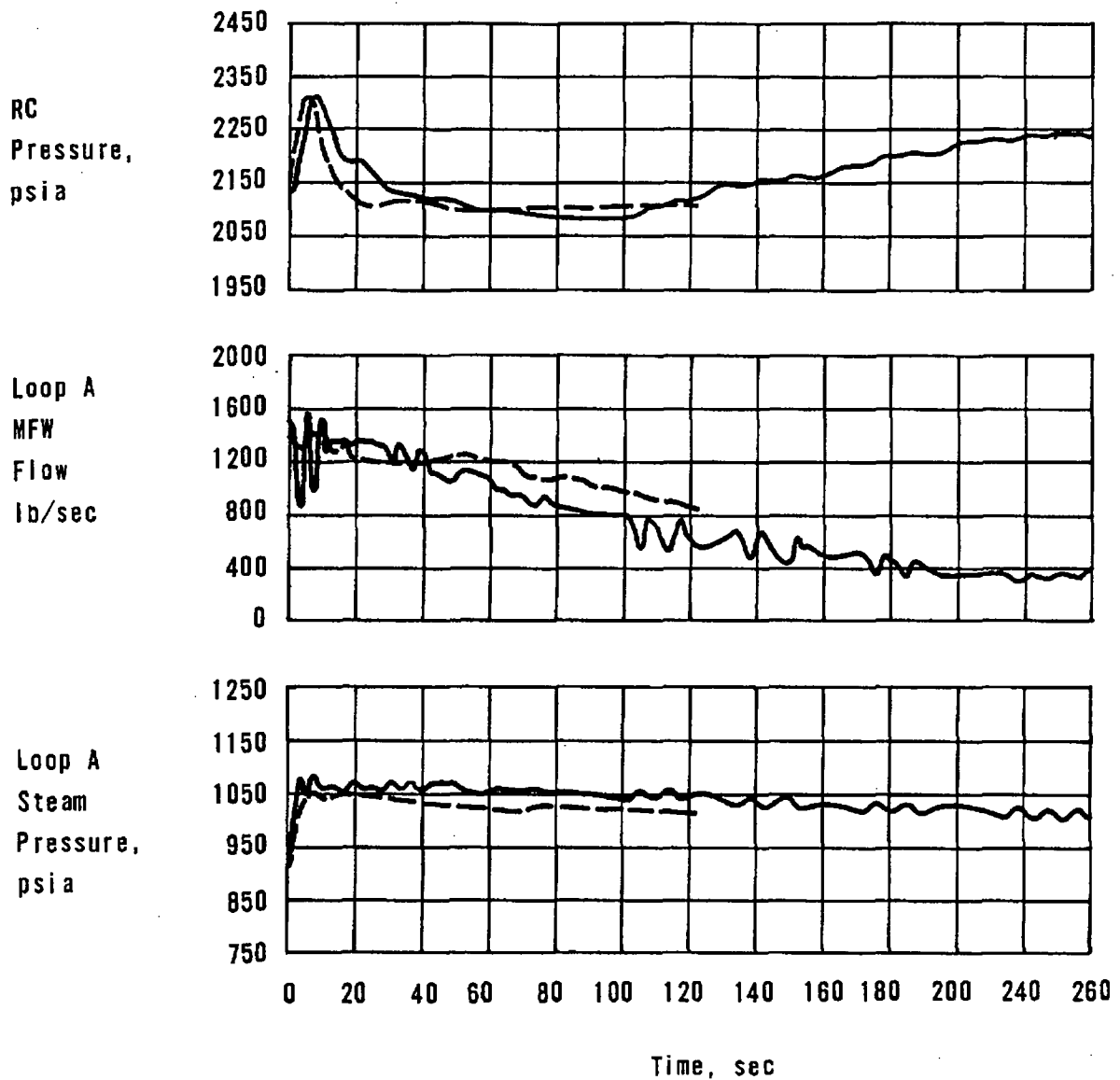


Figure 4-19. Plant Data Vs POWER TRAIN Data for  
One-RC-Pump Trip - 177-FA NSS

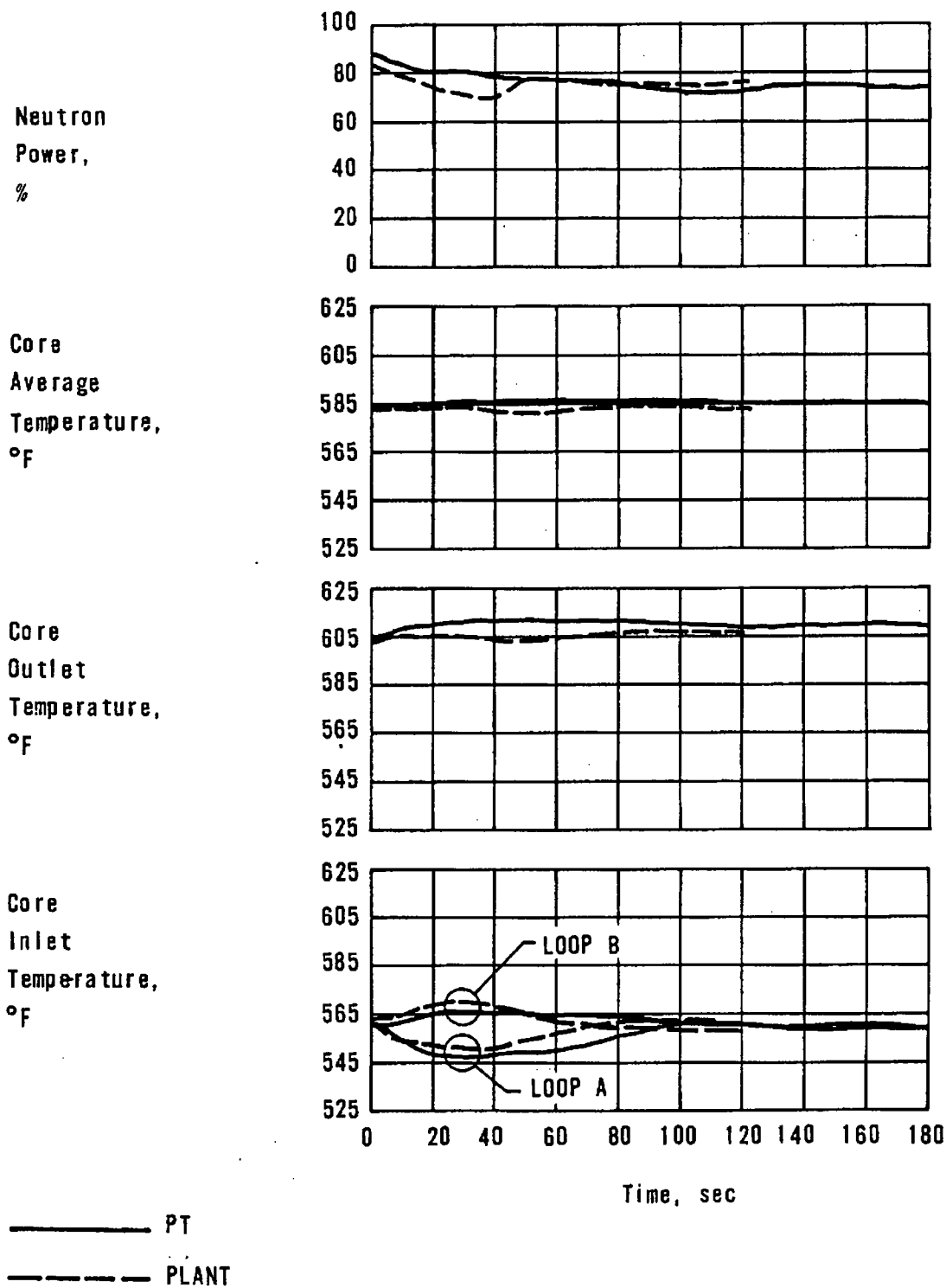
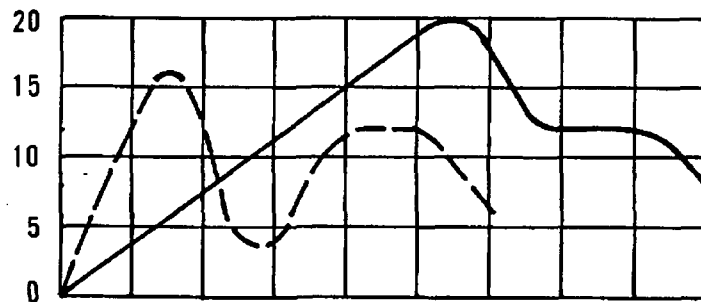


Figure 4-19. (Cont'd)

Change in  
Pressurizer  
Level During  
Transient, in.



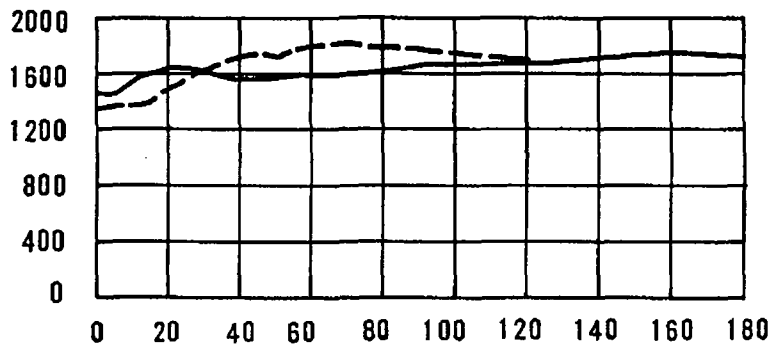
Core  
Outlet  
Pressure,  
psia



Loop A  
MFW Flow,  
lb/sec



Loop B  
MFW Flow,  
lb/sec



Time, sec

———— PT  
----- PLANT

## 5. REFERENCES

- <sup>1</sup> ISL - Interactive Simulation Language Programming Manual, Interactive Mini Systems, Kennewick, Washington (1974).
- <sup>2</sup> D. J. Bender, "An Explicit Unlimited Stability Approach to the Transient Conduction-Convection Equations," The Effective Use of Computers in the Nuclear Industry, USAEC Division of Technical Information, April 21-23 (1969) pp 561-585.
- <sup>3</sup> J. R. S. Thom, et al., "Boiling in Subcooled Water During Flow Up Heated Tubes or Annuli," Proc. Inst. Mech. Engrs., Vol 180, Part 3C (1965-1966)
- <sup>4</sup> N. Zuber, et al., Steady State Transient Void Fractions in Two-Phase Flow Systems, GEAP-5417, General Electric, January 1967, p 251.

**APPENDIX A**

**POWER TRAIN/CADD Current Model Comparison With  
FSAR Analysis for Consumers Power Midland Units**

## 1. Introduction

A Midland (Consumer's Power Company) FSAR Chapter 15 event that was analyzed using an earlier version of POWER TRAIN to provide input to CADD has been repeated with the current version of POWER TRAIN and CADD. The event is the reduction in main feedwater temperature at full power without control system action prior to the reactor trip.

The results of this POWER TRAIN-CADD event comparison indicates excellent agreement during the pre-trip portion of the event (power increase), and shows that the FSAR analysis is conservative with respect to reactor coolant temperature and pressure in the post-trip period.

## 2. Discussion

The event for which the comparison is made is an assumed reduction in main feedwater temperature to both steam generators at full power. No control system action (including feedwater control, reactor control, pressurizer heater, spray, or PORV actuation, occurs in the pre-trip period. The main feedwater temperature transient imposed is a reduction of 85F, assumed to occur over a 10-second period. The assumed feedwater temperature decrease causes (1) heat removal from the steam generators to increase, (2) reactor average temperature to decrease, and (3) reactor power to increase due to the assumed highly negative moderator reactivity feedback. The rate of change of reactor power is approximately 0.3%/second, so that the high flux trip setpoint is reached some 30 seconds after the feedwater temperature reduction occurs.

The analysis procedure for the Midland FSAR case and this case is as follows:

1. POWER TRAIN is initialized at full power with all automatic controls in manual or disabled. Physical constants correspond to end-of-cycle conditions.
2. A main feedwater temperature change of 85F is introduced by a manual switch.
3. The event progresses as shown in Figure A-1 until the reactor trips on high flux.
4. Upon reactor trip, the feedwater control and turbine bypass systems are assumed to operate. The feedwater control actions are to reduce main feedwater flow much more rapidly than normally expected to conservatively overpredict reactor coolant conditions, i.e., temperature and pressure.

5. Heat demand data (heat removal from the reactor coolant system by the steam generators) are obtained from POWER TRAIN and tabulated for input to CADD.
6. CADD is initialized to the same initial conditions and time in core cycle as POWER TRAIN. The course of the event is calculated by CADD.

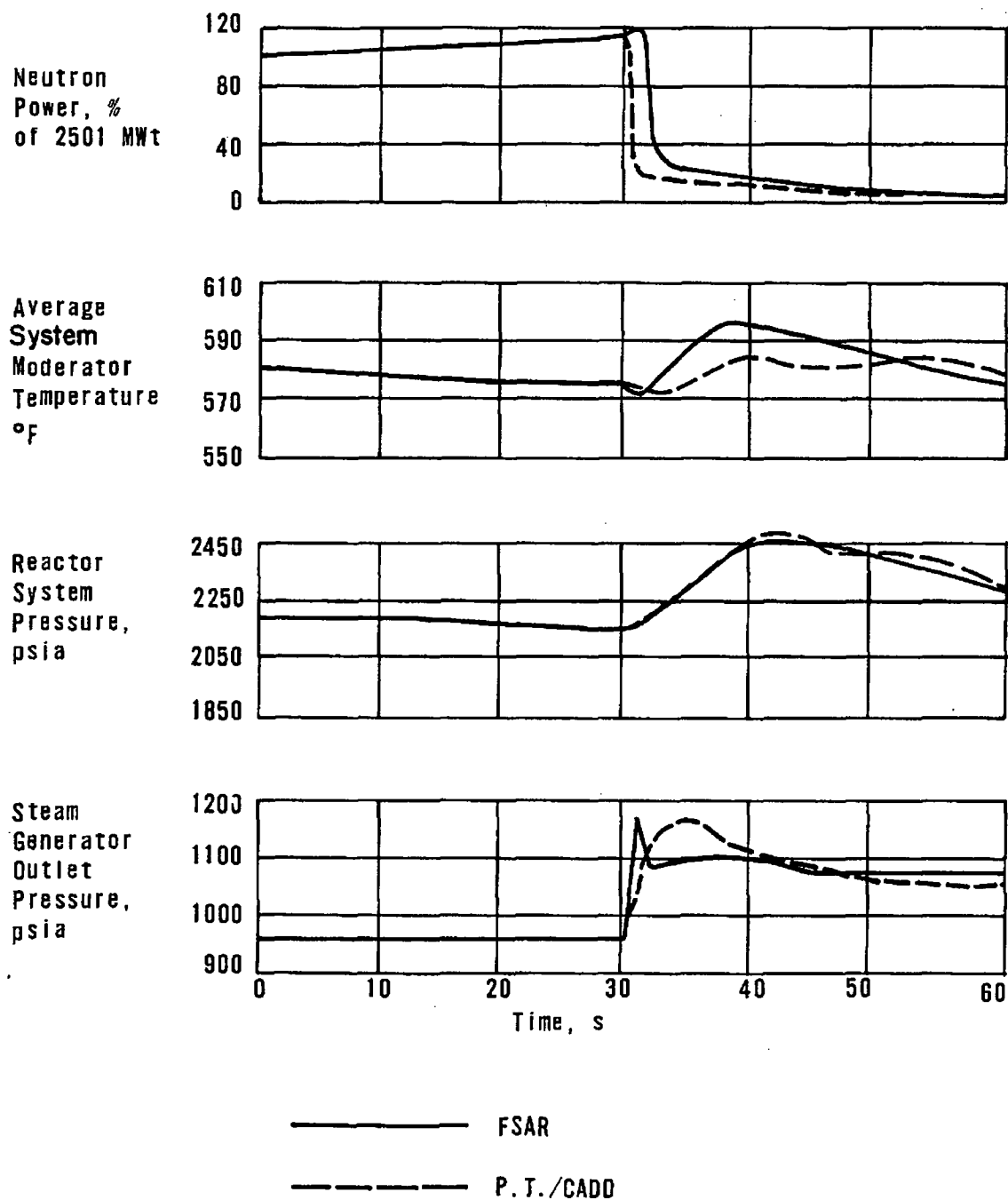
The data that describe the course of the event were derived as described above and are shown in Figure A-1.

### 3. Results

The current analysis with POWER TRAIN and CADD indicates repeatable-performance reactor power and reactor coolant temperature and pressure during the pre-trip phase of the event.

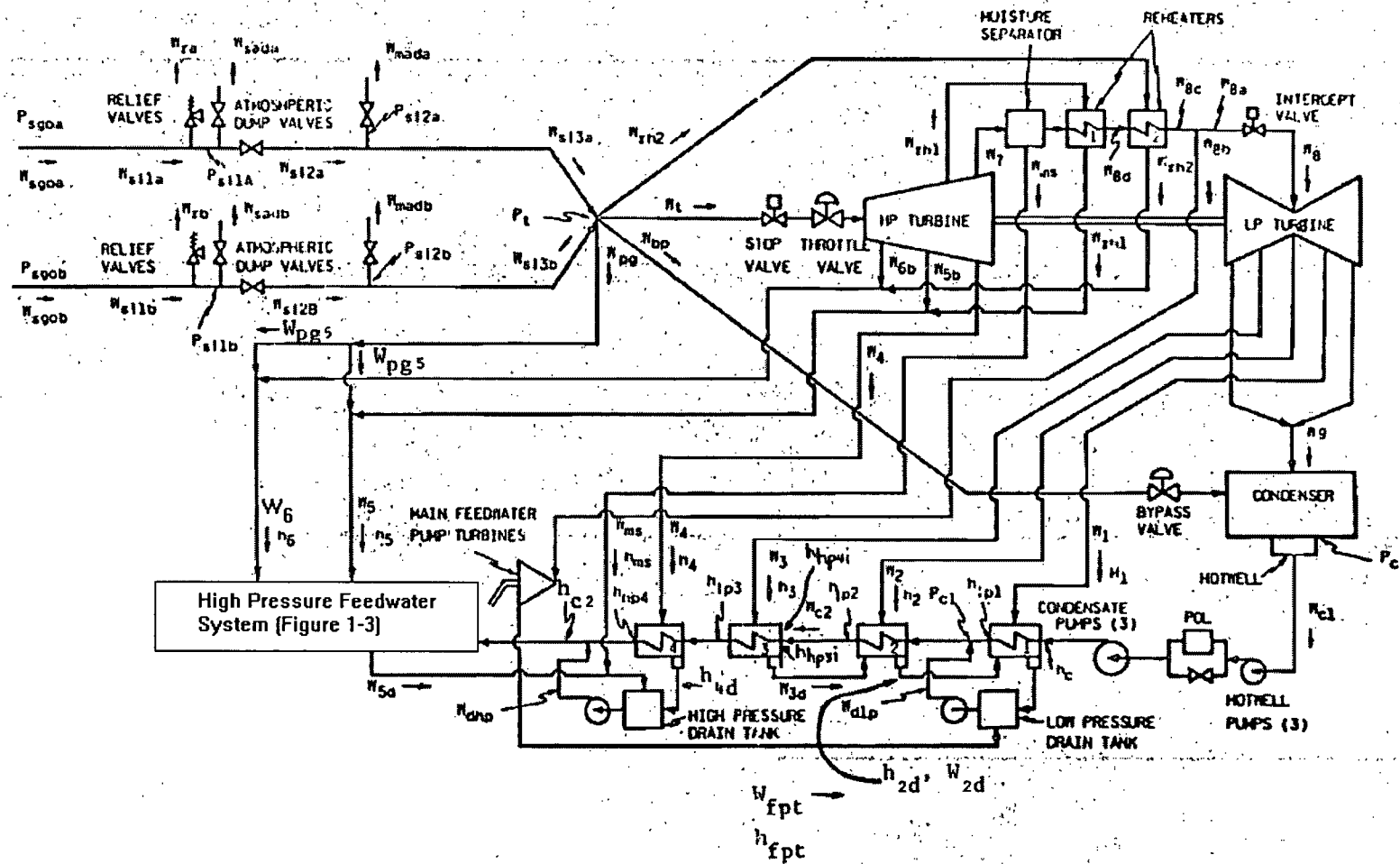
In the post-trip period, the major variables are equal to or less than the FSAR analysis (e.g., RC pressure). As a result, the current models (POWER TRAIN and CADD) demonstrate agreement with the FSAR analysis to the time of trip, and the FSAR analysis is conservative relative to the current models.

Figure A-1. Midland FSAR Event Analysis — Decrease of 85F in Main Feedwater Temperature



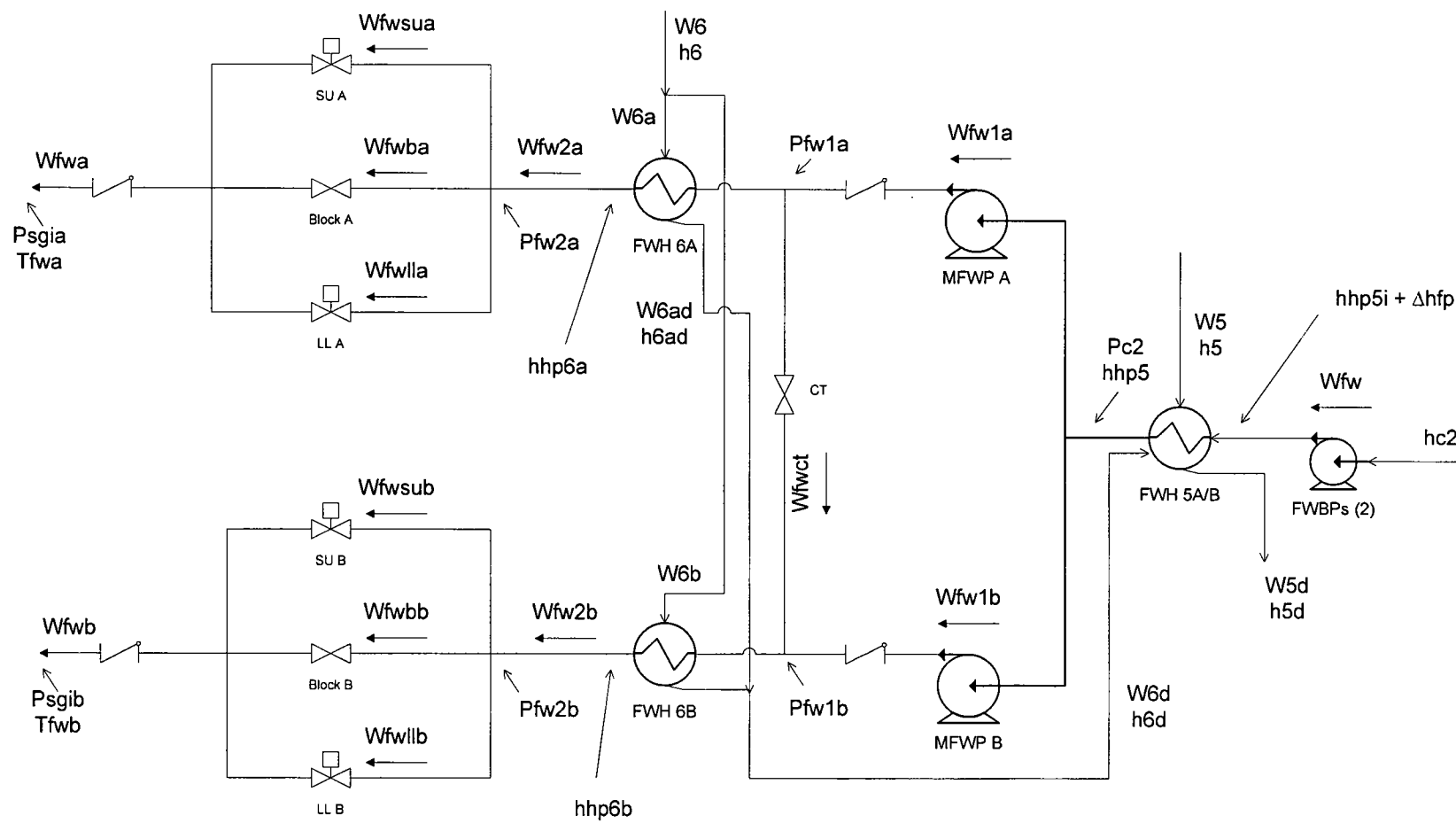
# CR-3 Secondary Plant Layout for DPT

Figure 1-2: Secondary System Diagram



### CR-3 Secondary Plant Layout for DPT

### Figure 1-3: High Pressure Feedwater System



**FLORIDA POWER CORPORATION**

**CRYSTAL RIVER UNIT 3**

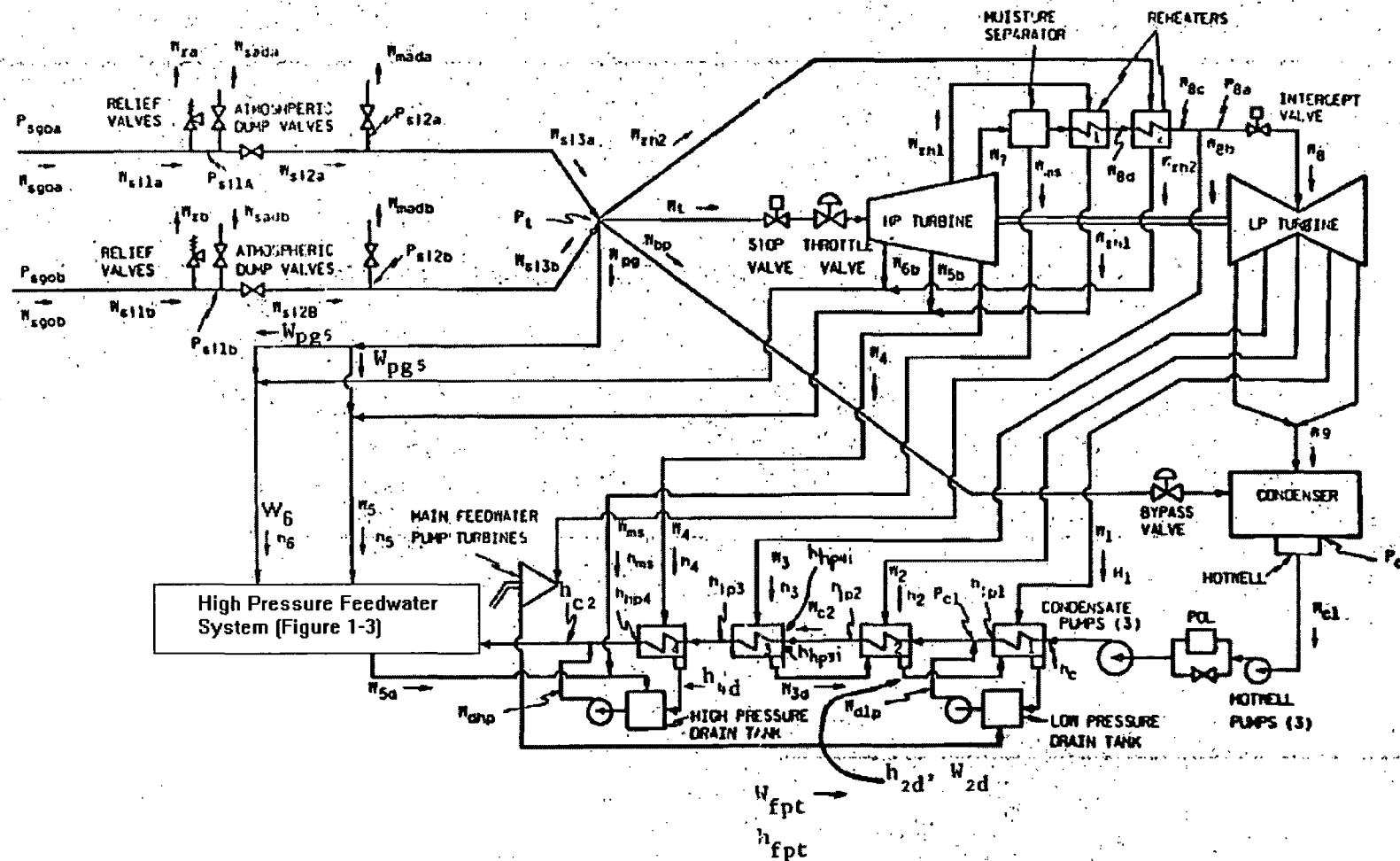
**DOCKET NUMBER 50-302 / LICENSE NUMBER DPR-72**

**ENCLOSURE 2**

**DIGITAL POWER TRAIN SECONDARY SYSTEM DIAGRAMS  
FOR CR-3**

### CR-3 Secondary Plant Layout for DPT

### Figure 1-2: Secondary System Diagram



**Figure 1-3: High Pressure Feedwater System**

